AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

Vol. 44 No. 6 JUNE 1954 PRICE: 3s. 6d.

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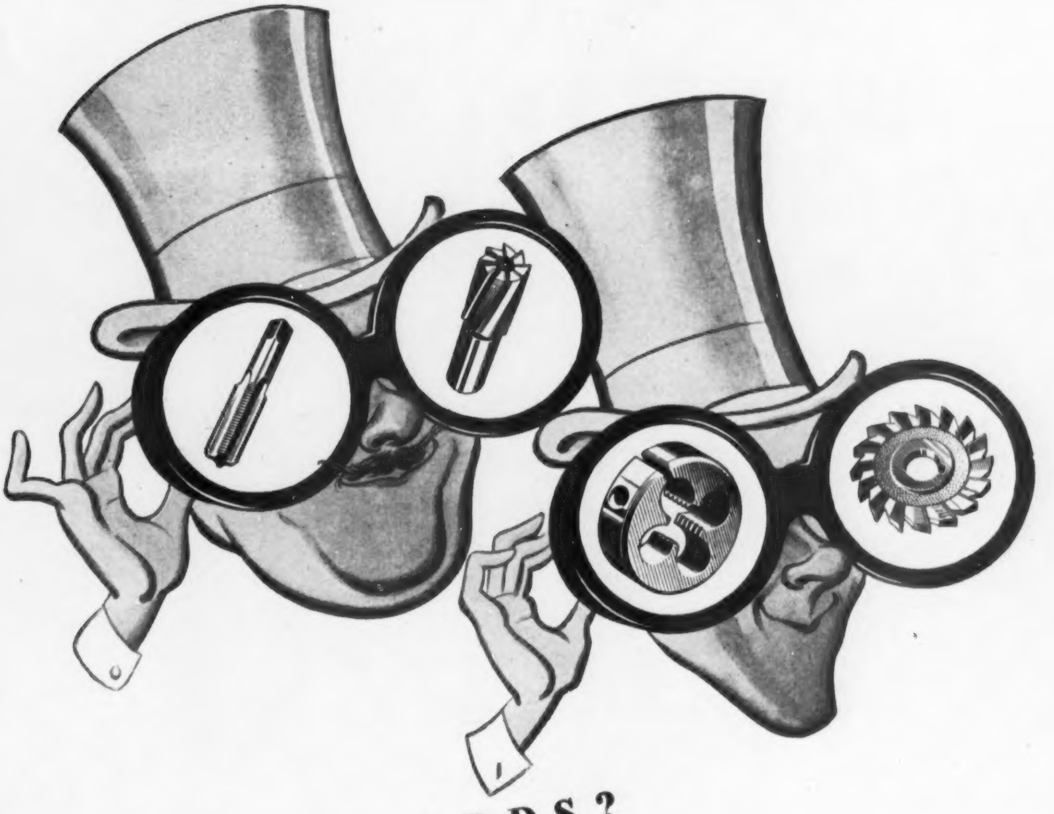

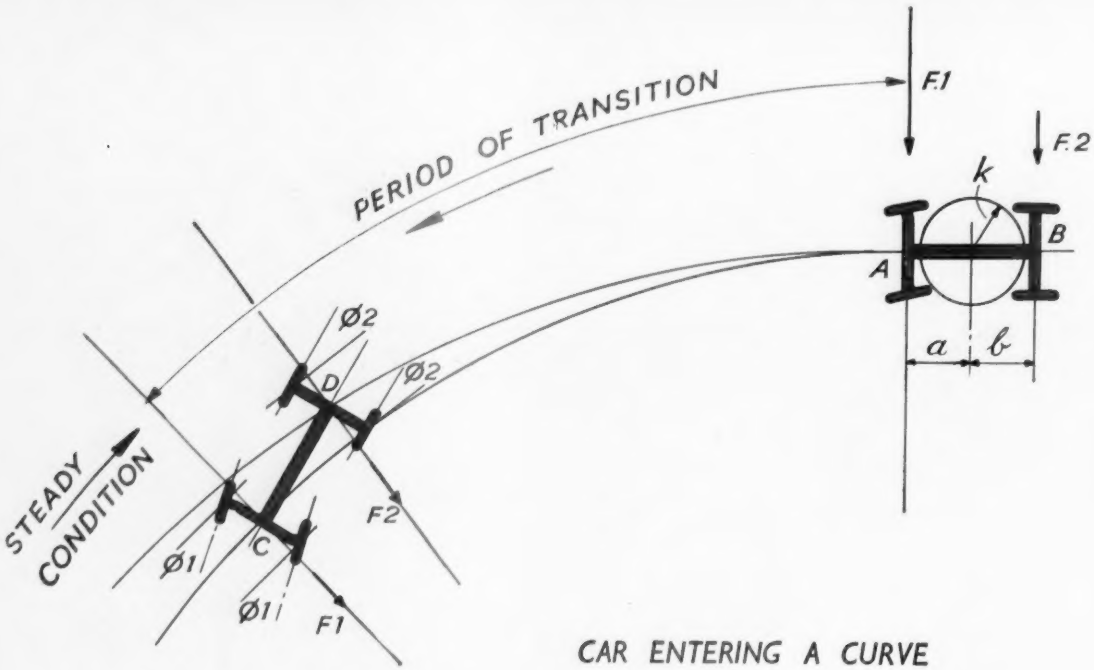

CAR ENTERING A CURVE

SUBTLETIES OF STEERING

The Transient Stage: 2. K^2/ab Ratio

As we have seen, the transient stage, in car steering or handling behaviour, occurs between any two adjacent steady conditions. For the sake of simplicity it is easiest to consider the transient stage between straight ahead and a curve of given radius. For the same reason, it is easiest to consider the front wheels moved over instantaneously to the lock angle appropriate to the steady curve which it is desired to achieve. This instantaneous locking over of the front wheels causes the front tyres to develop a drift angle and hence a sideways force which pushes the front of the car inwards and urges it gradually onto the curved path which it is desired to take. This gives the car an angular acceleration about a vertical axis.

Now if the moment of inertia of the car about a vertical axis through its centre of gravity is such that K^2/ab is unity (K being its radius of gyration and a and b respectively the distances of the front and rear wheels from the centre of gravity) the vehicle can be represented by masses W_f and W_r, equal to the weights carried by front and rear wheels, placed at those wheel centres. In this case the centripetal acceleration of the front end of the car due to the drift angle produced as described above produces no reaction at the rear wheels, and the sideways reaction at the rear will only build up gradually as the car goes onto a curved path.

If we have the much more common condition where the K^2/ab ratio is less than unity, then the equivalent masses W_f and W_r will lie within the wheelbase, and the immediate effect of the angular acceleration is to induce an inward force at the rear wheels, which will then as before gradually build up to the final steady value. In both these cases the force at the rear wheels during the transient stage is in one direction only.

Now consider the possible but very rare case of K^2/ab being greater than unity. To represent the vehicle by the two masses W_f and W_r once more, they must now lie beyond the wheelbase. Because of this, or if you prefer it basically because the K^2/ab ratio is greater than unity, the inward force at the front wheels must be greater than the centrifugal force, and there must therefore be an initial outward force at the rear wheels. As sideways acceleration builds up, this will demand an eventual inward force at the rear wheels, and there will therefore be a change of direction of the force at the rear wheels during the transient stage.

This change of direction of the force at the rear wheels implies a good deal of 'float' there and a resulting feeling of instability at the rear end of the vehicle.

At one time, shortly after the general introduction of independent front suspension in America in 1934, there was a tendency to make the K^2/ab ratio as large as possible in order to equalize the pitch and bounce frequencies of the car, and so avoid interference 'kicks' in the ride. If this tendency was taken too far in certain cases, and occurred on an over-steering car (which demands more frequent changes of course and on which the back has to swing out further anyhow) it would be responsible for the often-expressed argument that cars with too high a K^2/ab ratio were pigs to steer, and that you could have either a good ride or good steering, but not both.

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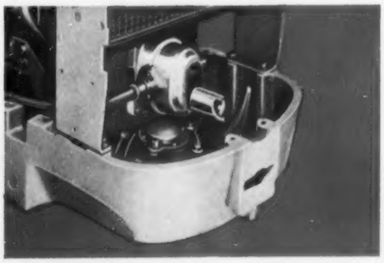
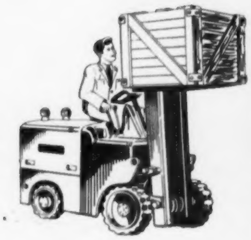
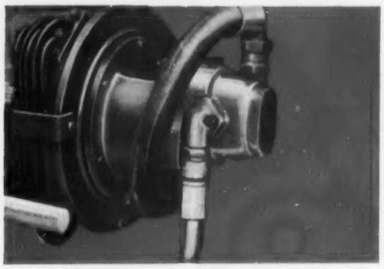


TRACTORS OF ALL TYPES

Many tractors of British manufacture incorporate the Plessey Hydraulic Pump for the operation of the power lift. The example illustrated is the "Massey Harris" 744D Tractor.





HEAVY LIFT TRUCKS

Note how easily the Plessey Pump has been introduced into the design of "Stacatruc" fork lift trucks. The Pump is here directly coupled to the motor unit.

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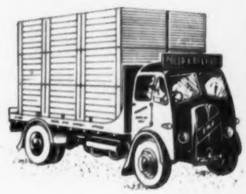

HEAVY TRANSPORT BRAKING

Here again, the adaptability of the pump is much in evidence. Mounted directly on the gear box of this E.R.F. Lorry it forms the heart of the hydraulic system giving pressure servo assistance for efficient braking under heavy load conditions.

* *Write for full information in Publication No. 680*

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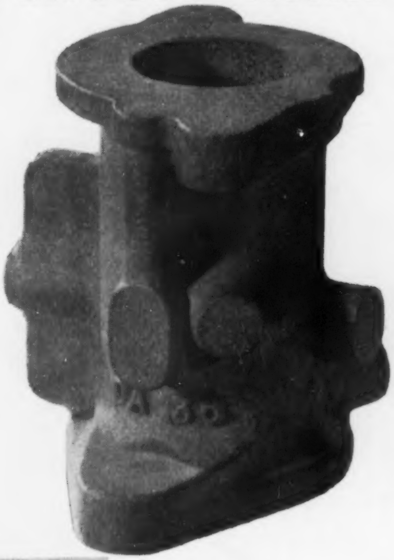

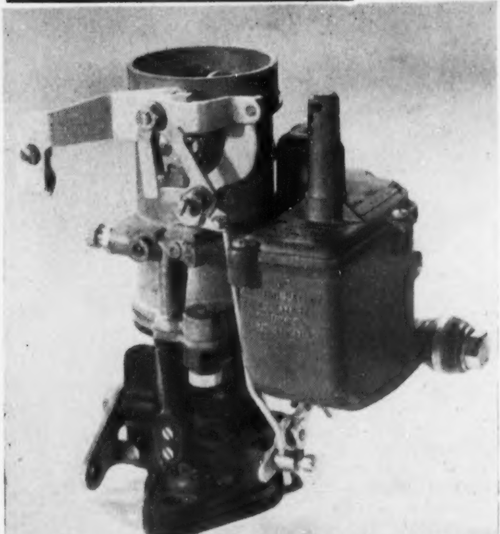

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THE INS AND OUTS

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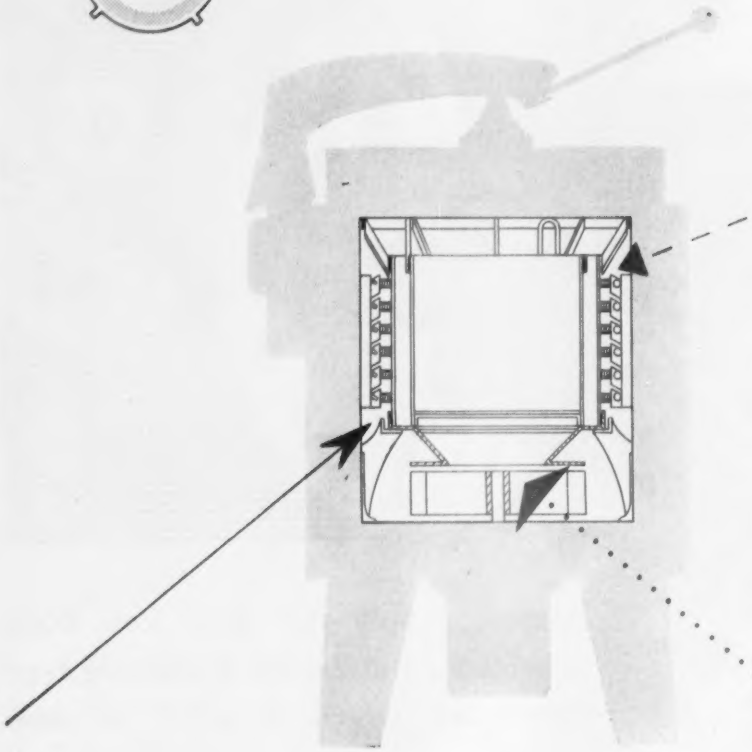

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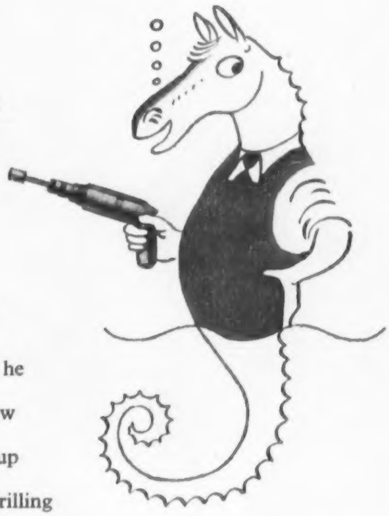

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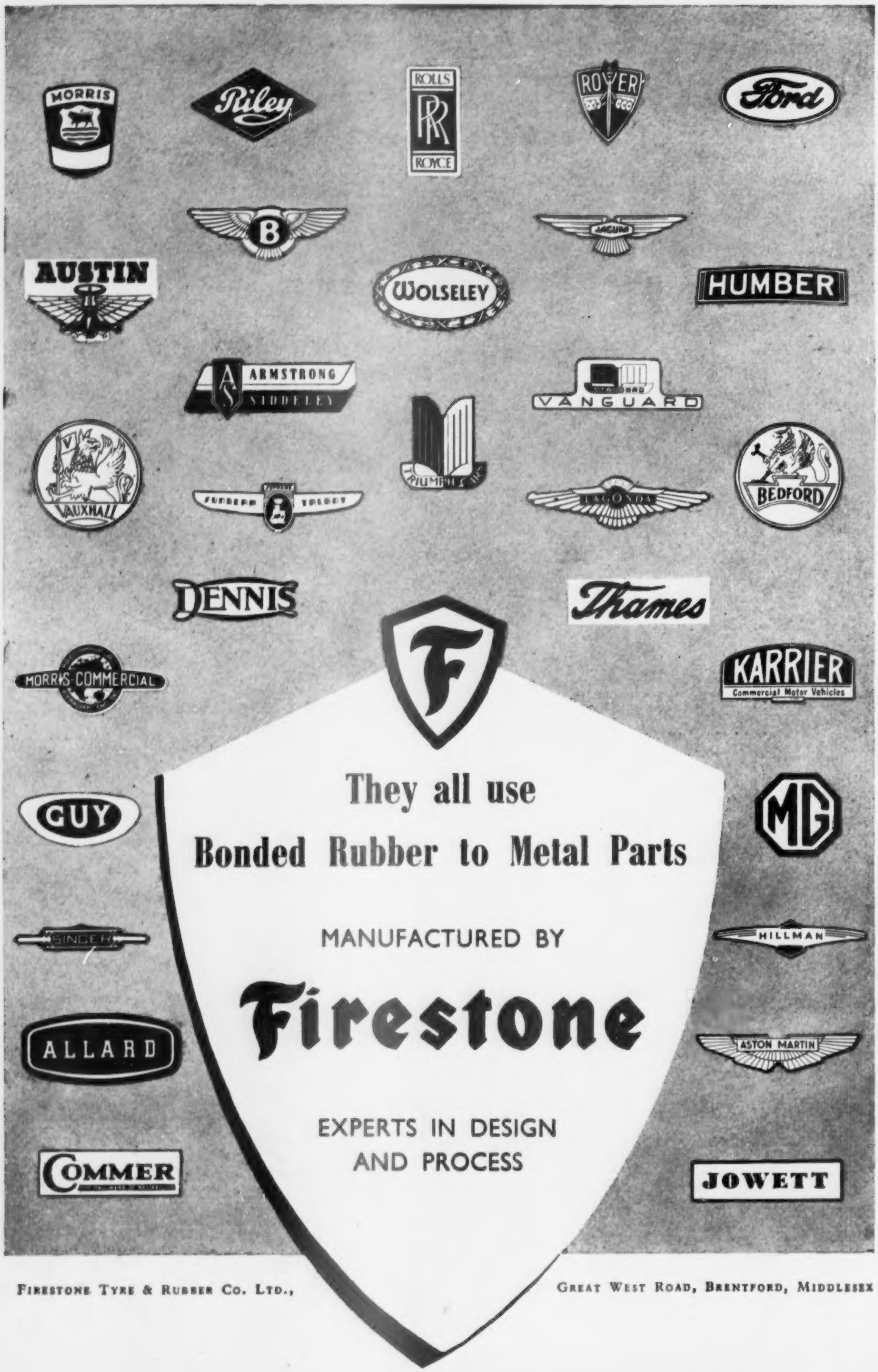
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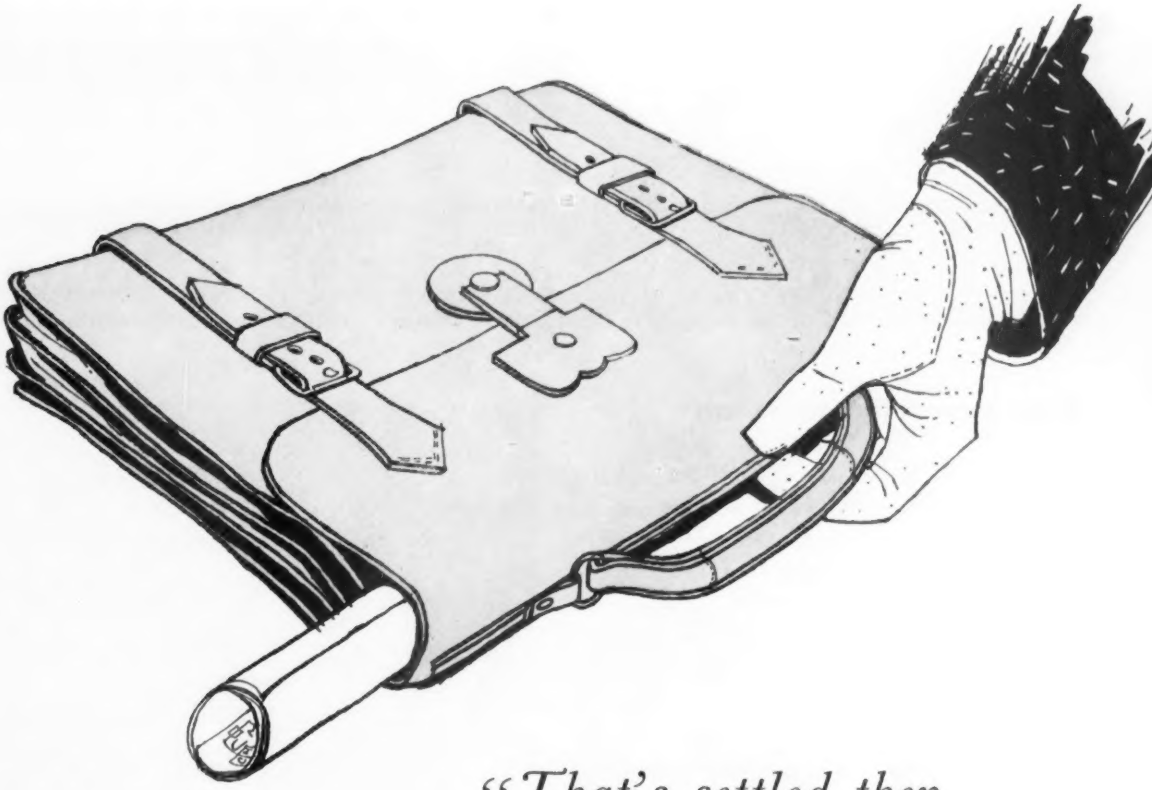

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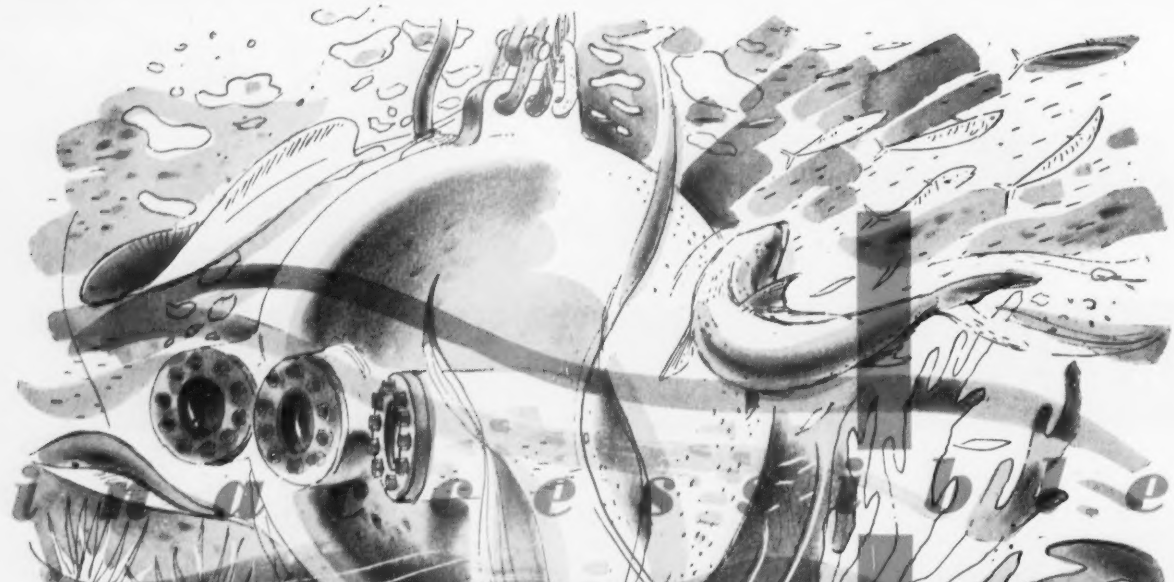

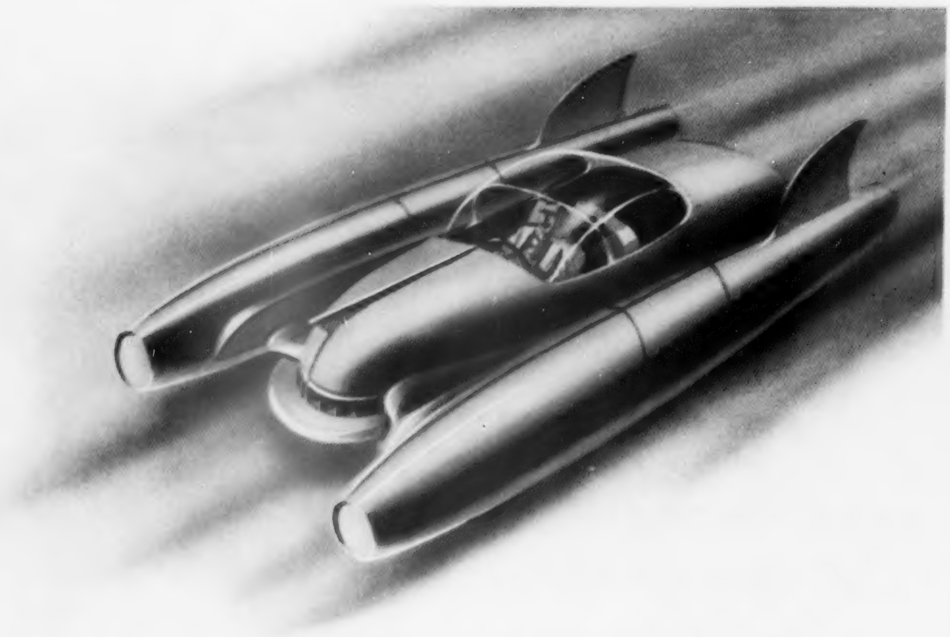


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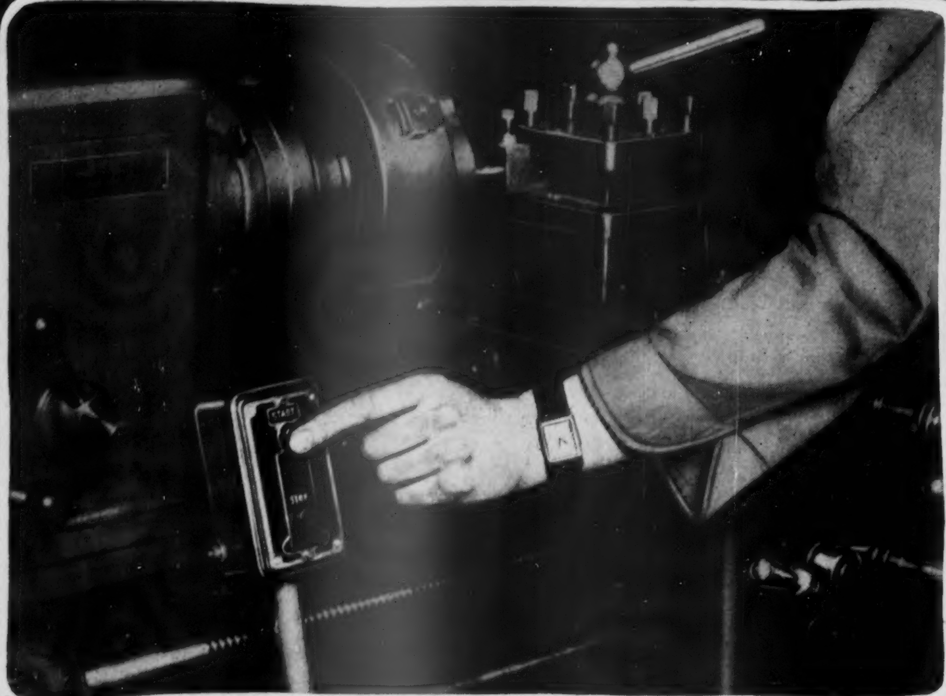

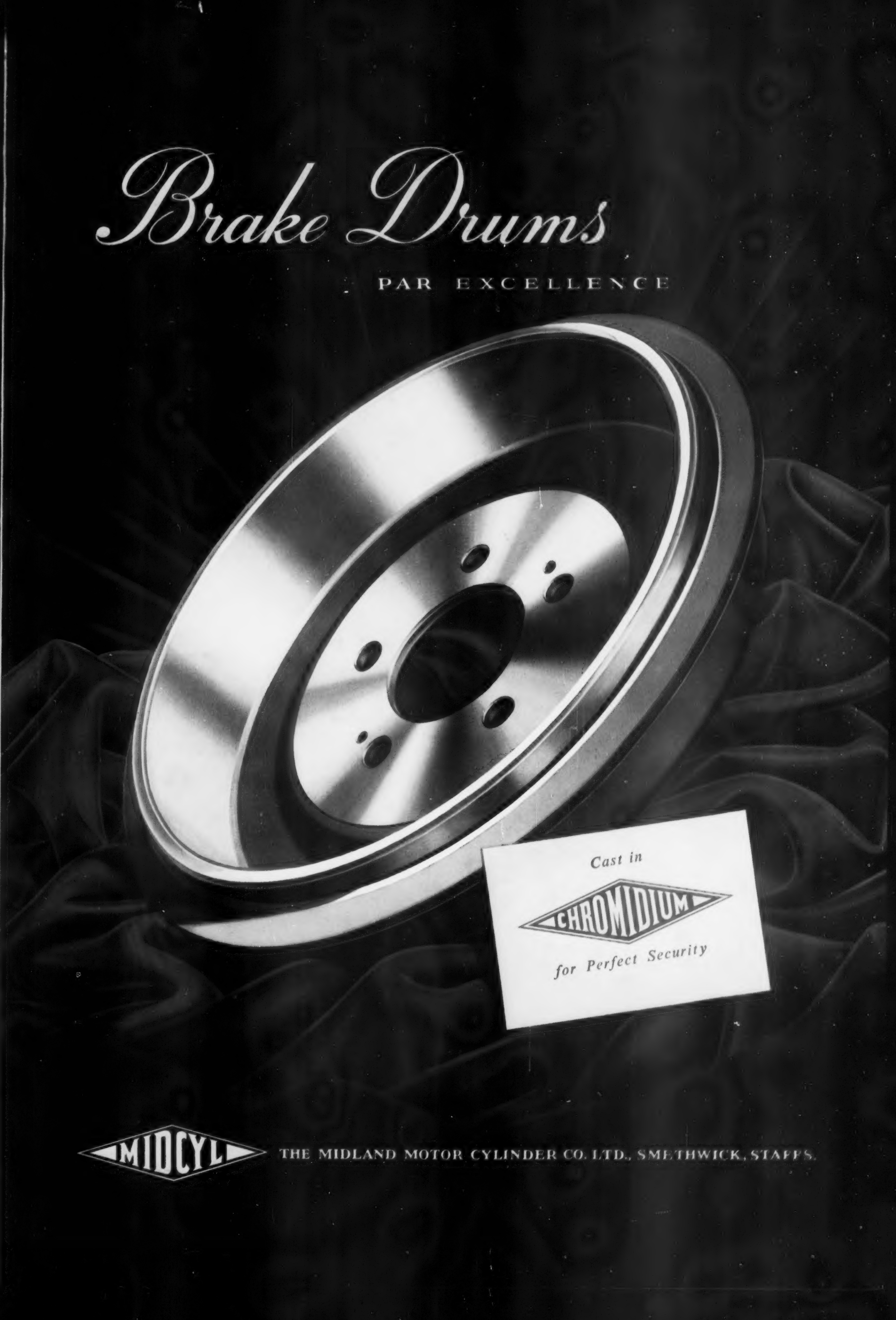
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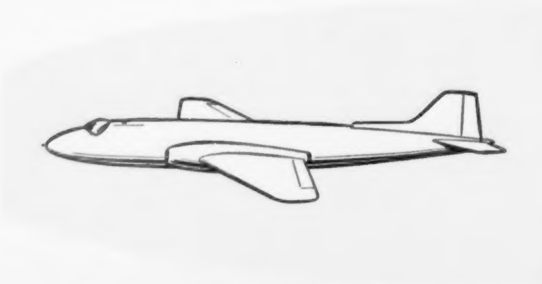

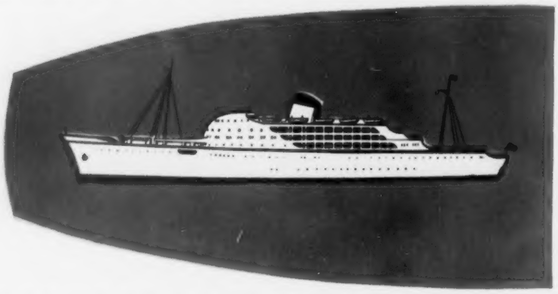

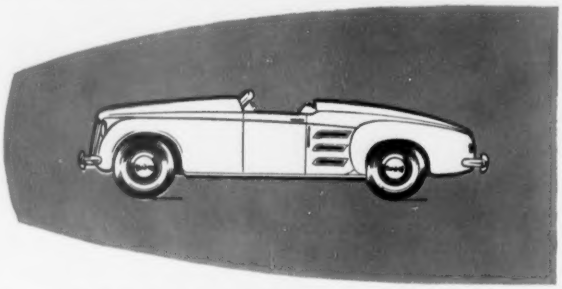

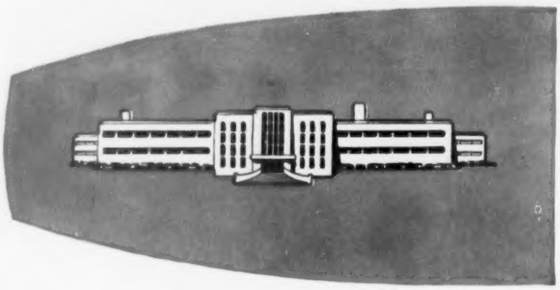

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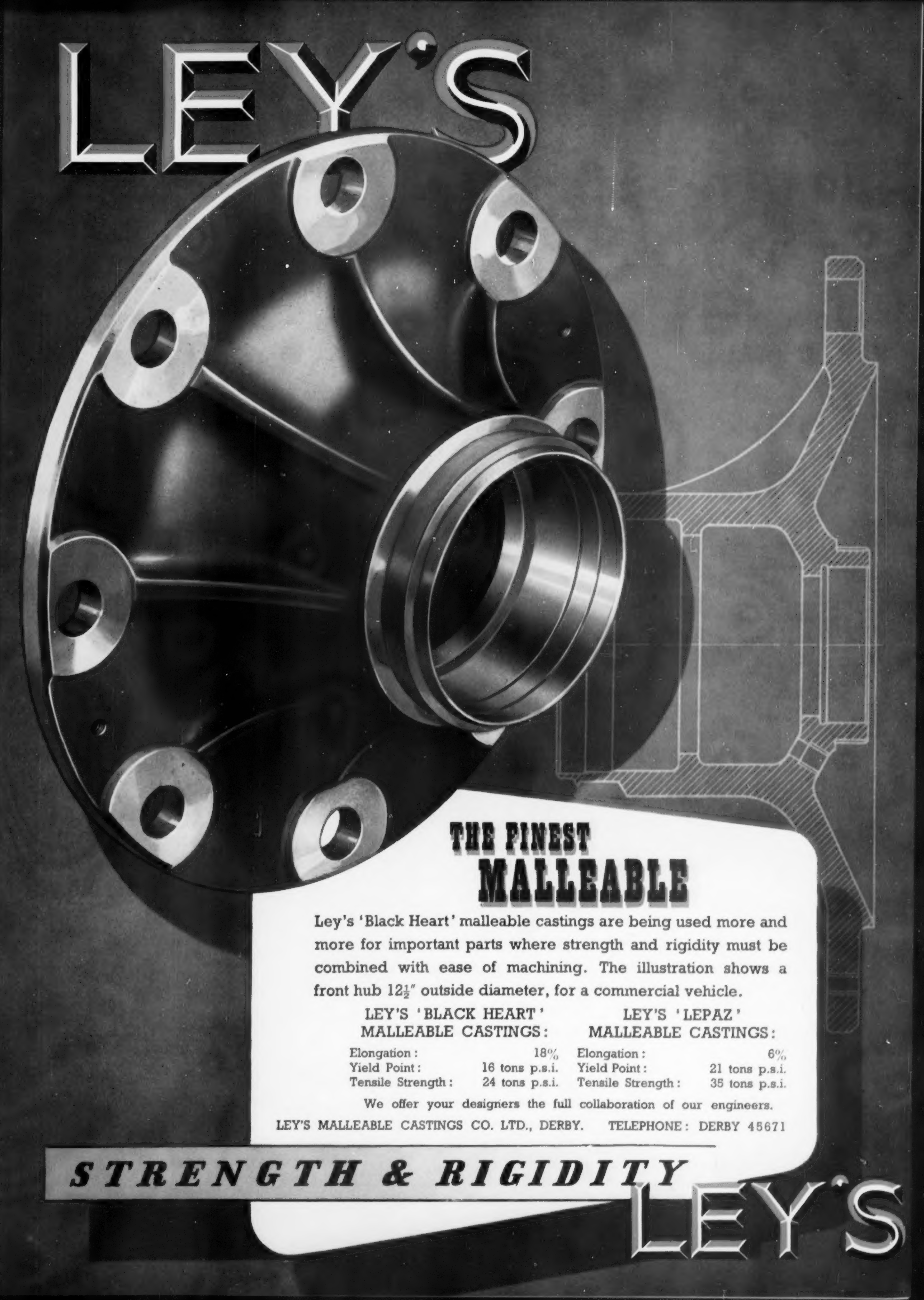
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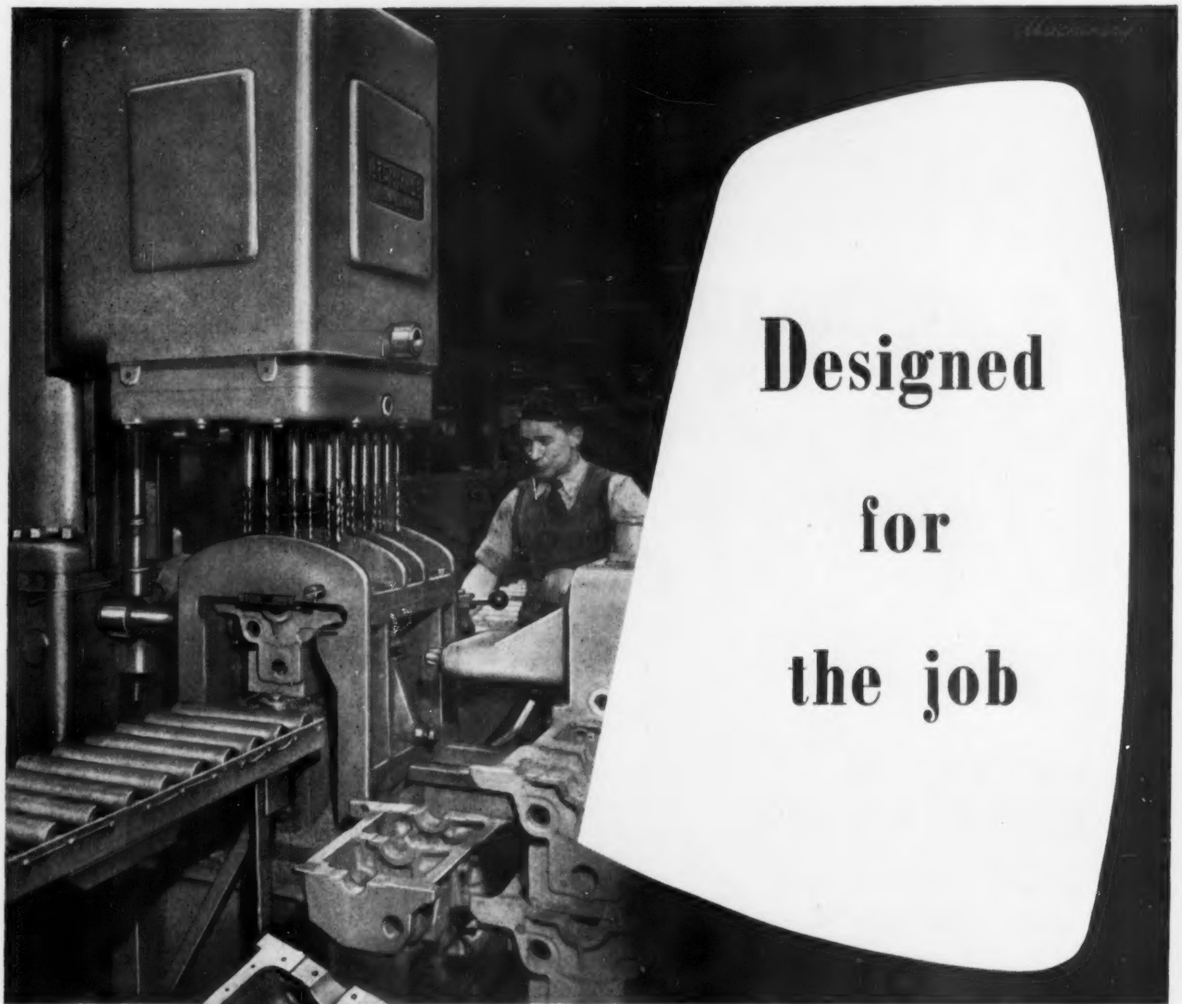

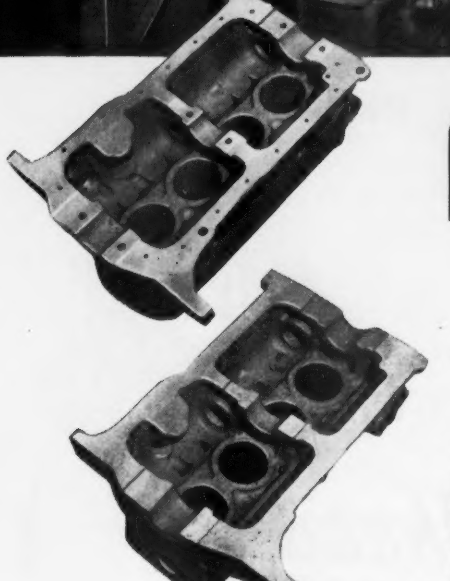

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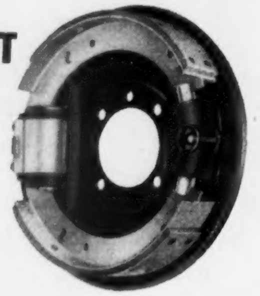

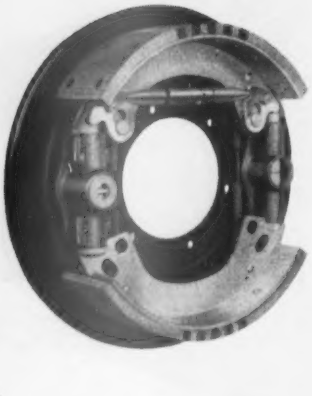

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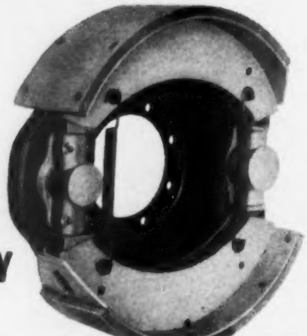

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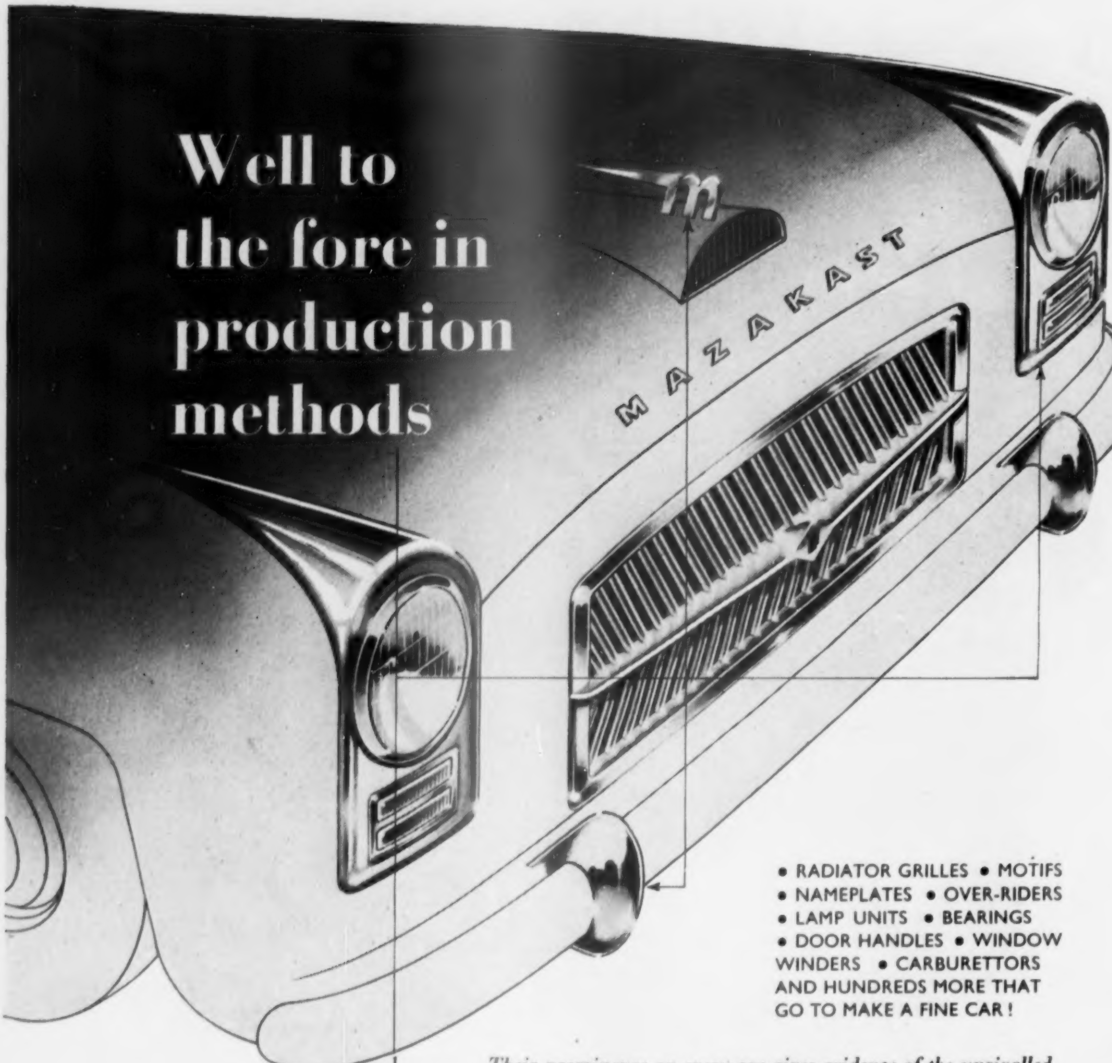

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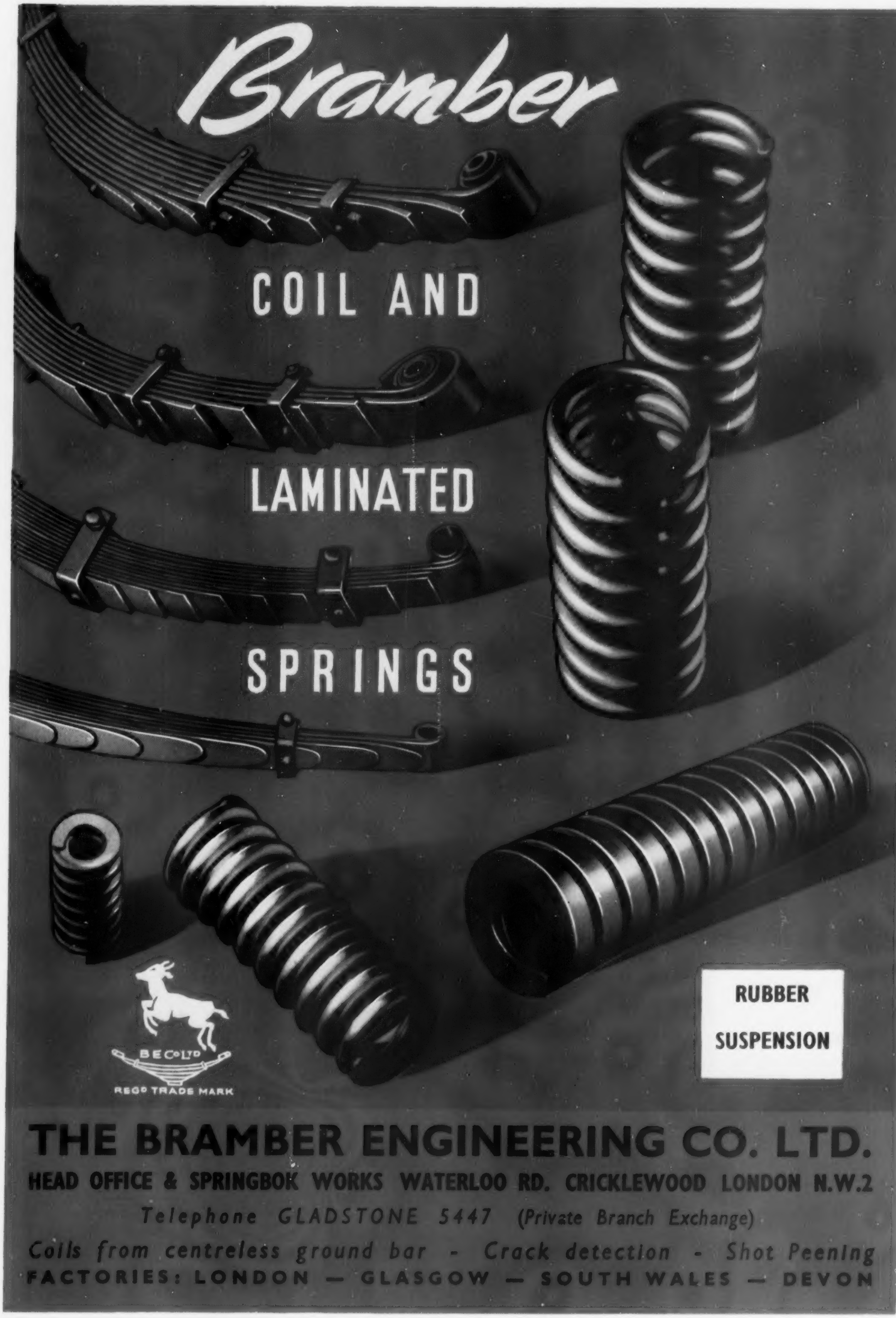
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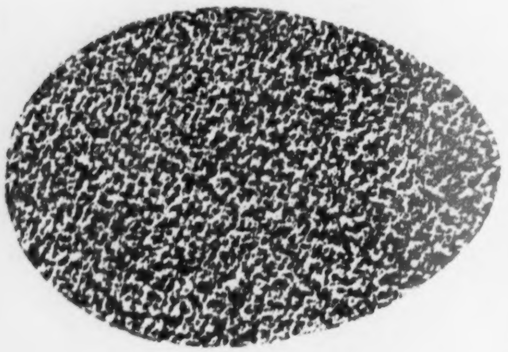

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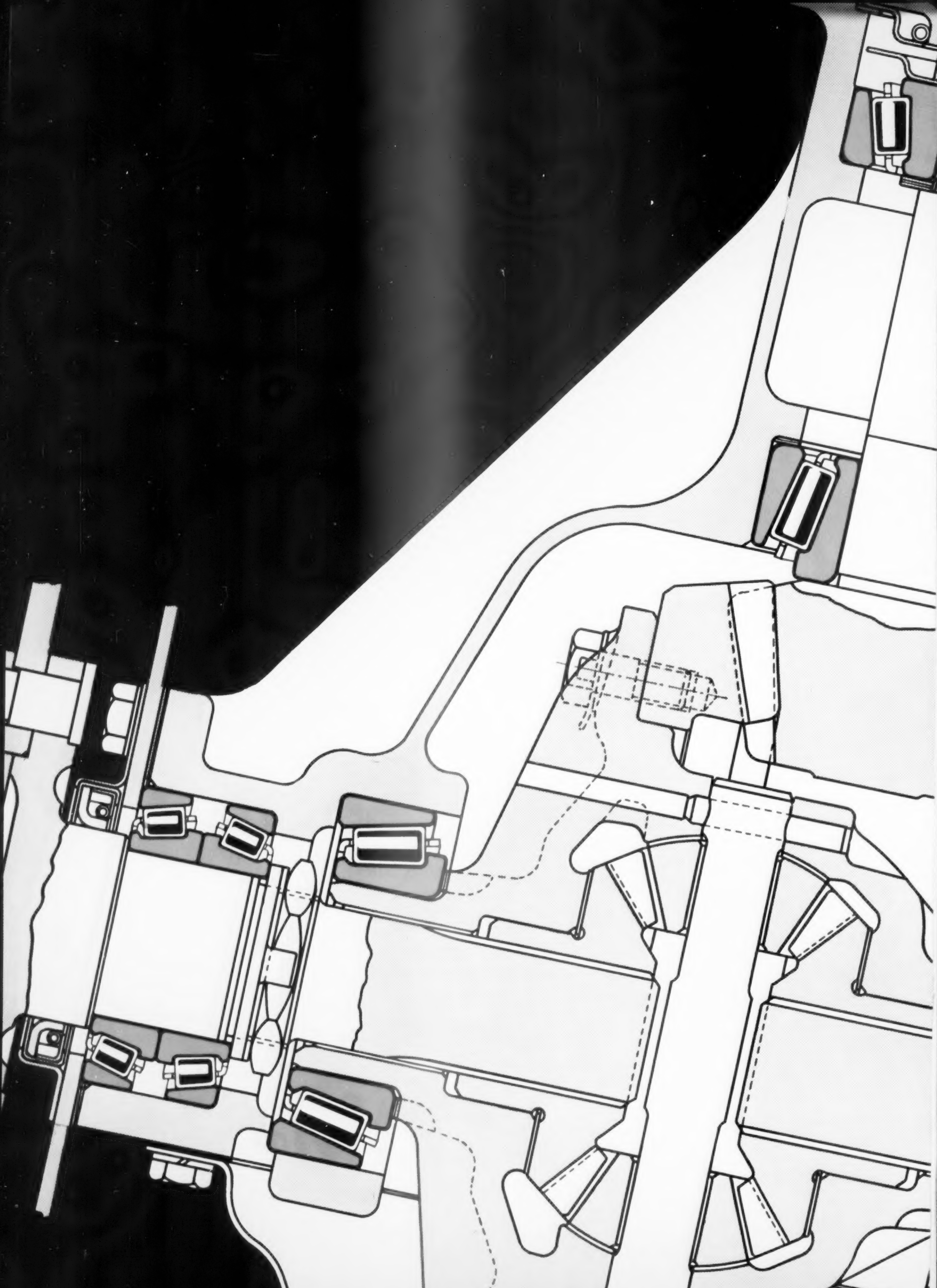

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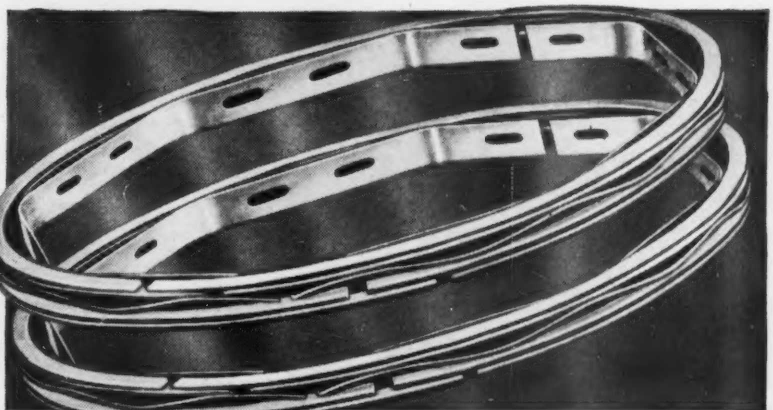

ALL over the country more and more fleet owners are trying Duaflex Oil Control Rings — and having tried them on one vehicle, are fitting them to the whole fleet. They find that Duaflex Rings do defer the need for a re-bore — and for a long time too! That means that their vehicles spend more time on the road — less time in the garage.

More economical operation

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Rebore results for pounds less

Provided the engine wear is not so great that a major overhaul is already necessary, a set of Duaflex oil control rings will do the job equally well at a fraction of the cost and a re-bore can be indefinitely postponed. Duaflex rings are designed to take up cylinder wear by sealing the gap between the piston and cylinder wall. They are self-compensating and adjust themselves automatically to irregularities in the bore. What's more, they are designed to reduce wear on cylinder walls — and the rings themselves last longer too!

100,000 MILES WITHOUT A RE-BORE

Duaflex Oil Control Rings have been given a severe testing in vans of the "Birmingham Post and Mail". Since April, 1948, twenty-six vans have been fitted with Duaflex Rings. Between them, these vans have completed 1,545,000 miles — without a re-bore!

One van has now exceeded 100,000 miles, five others are approaching the 95,000 mark.

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Mr. J. Jennings, who is responsible for the maintenance of the "Birmingham Post and Mail" fleet, is delighted with the results obtained with Duaflex. He says:

" . . . . . *Newspaper delivery vans are notoriously hard-worked, but Duaflex have stood up wonderfully well to continuous stop-start during driving and show a great saving in maintenance time and costs.*"

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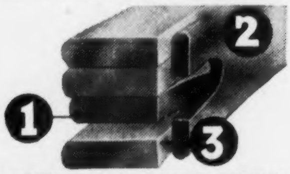
The principle of Duaflex rings is unique. They are designed to expand in two directions; vertically, to seal the rings in their grooves; and outwardly, to press against the cylinder walls. In this way an oil and gas-tight seal is maintained.



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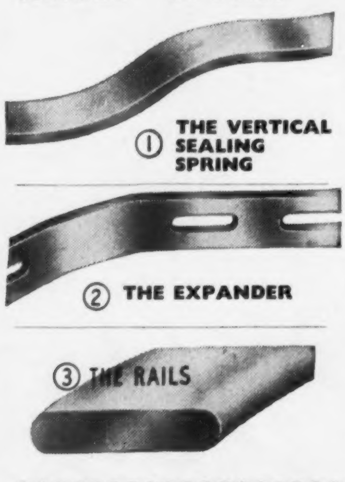

The vertical sealing spring (1) seals the ring in its groove. The "Expander" (2) maintains an even outward pressure ensuring perfect contact with cylinder walls however worn or distorted the bore may have become. And the specially shaped rails (3) are designed to 'wipe' oil from the cylinder walls and avoid scraping and consequent wear.

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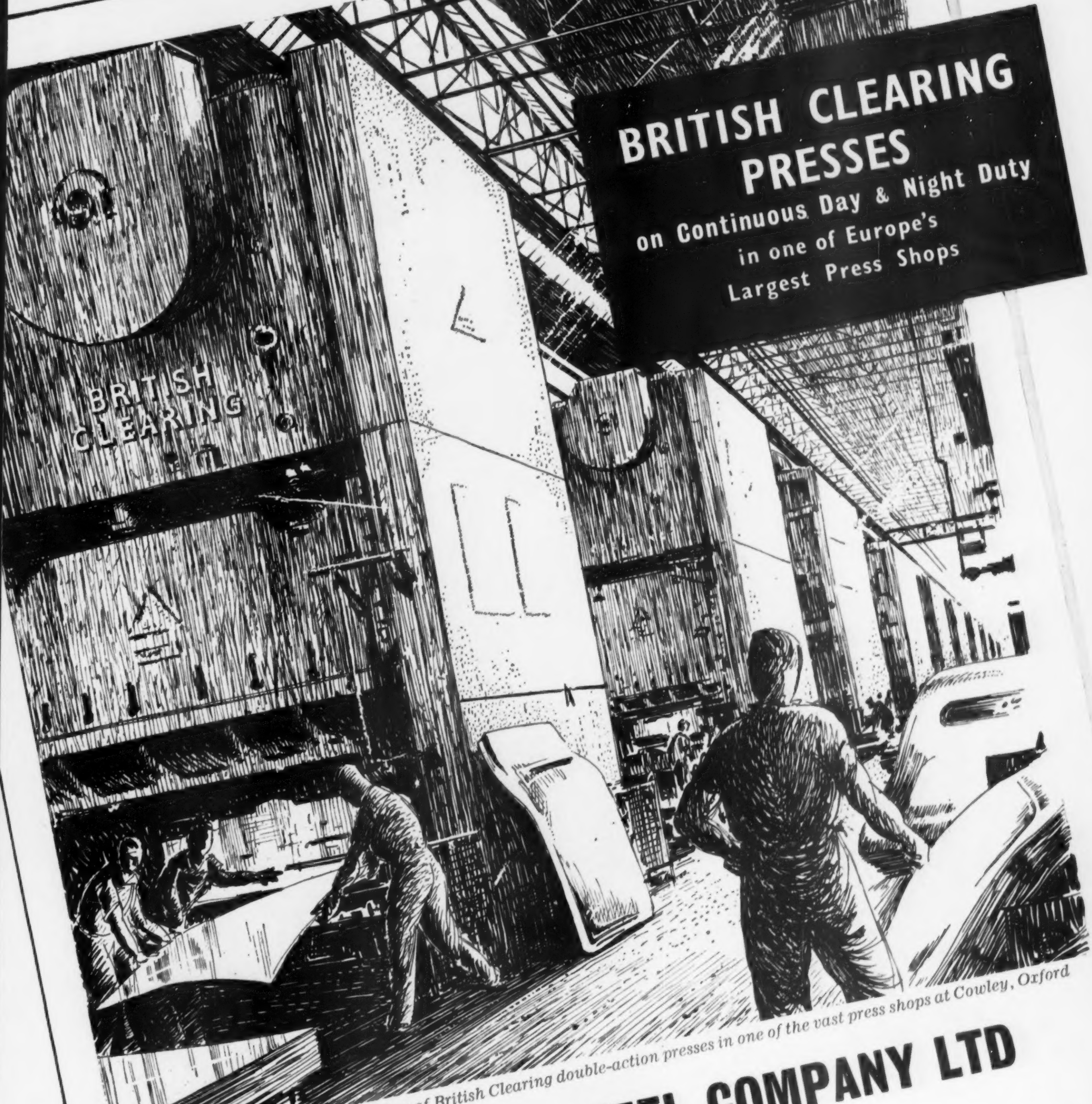
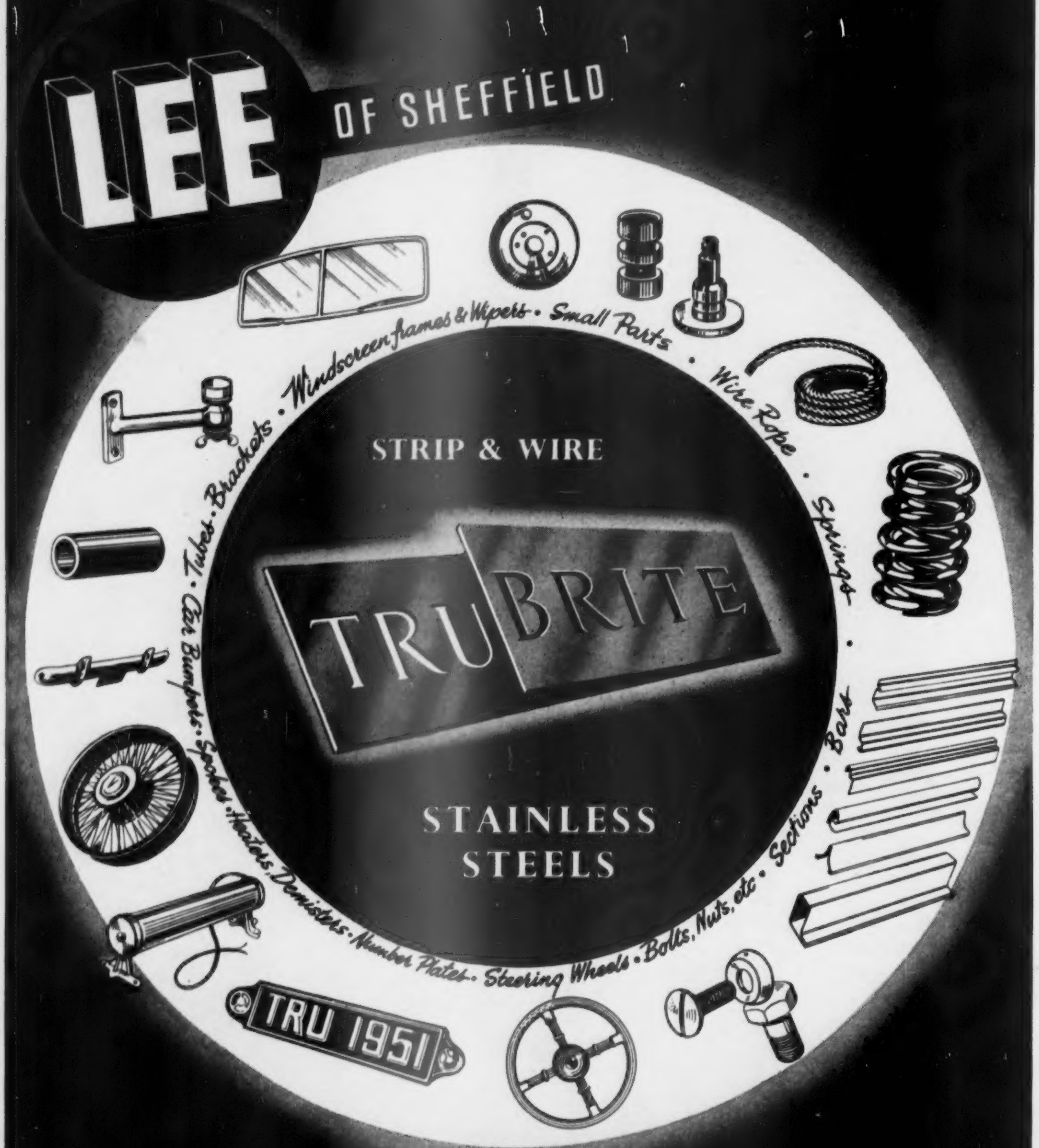


An artist's impression of a battery of British Clearing double-action presses in one of the vast press shops at Cowley, Oxford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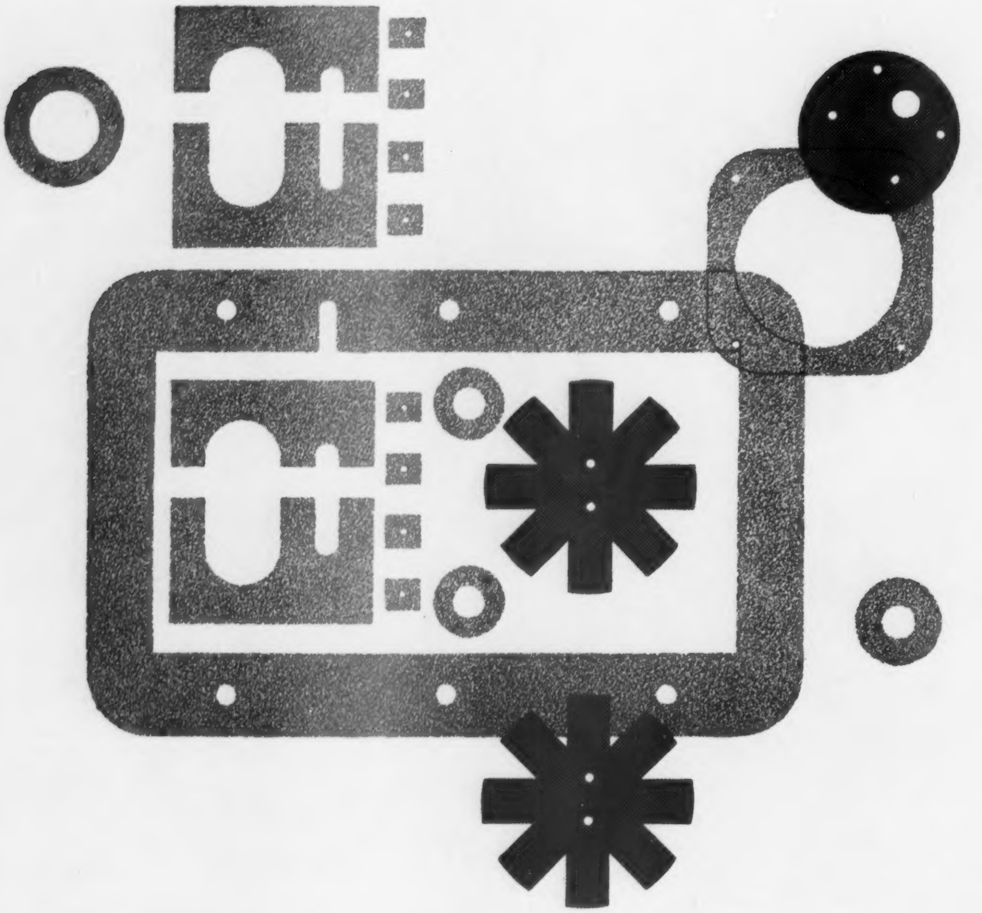


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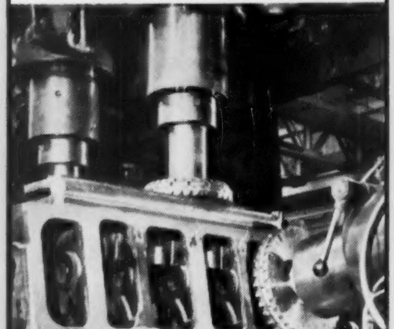

At Ruston & Hornsby Ltd., Lincoln, Wimet type 'A' cutters are machining diesel engine cylinder blocks on four faces simultaneously, with 20 h.p. applied to each spindle.
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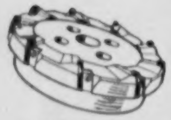

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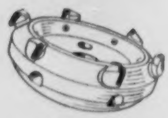

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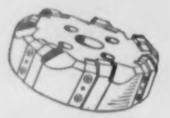

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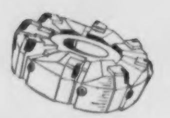

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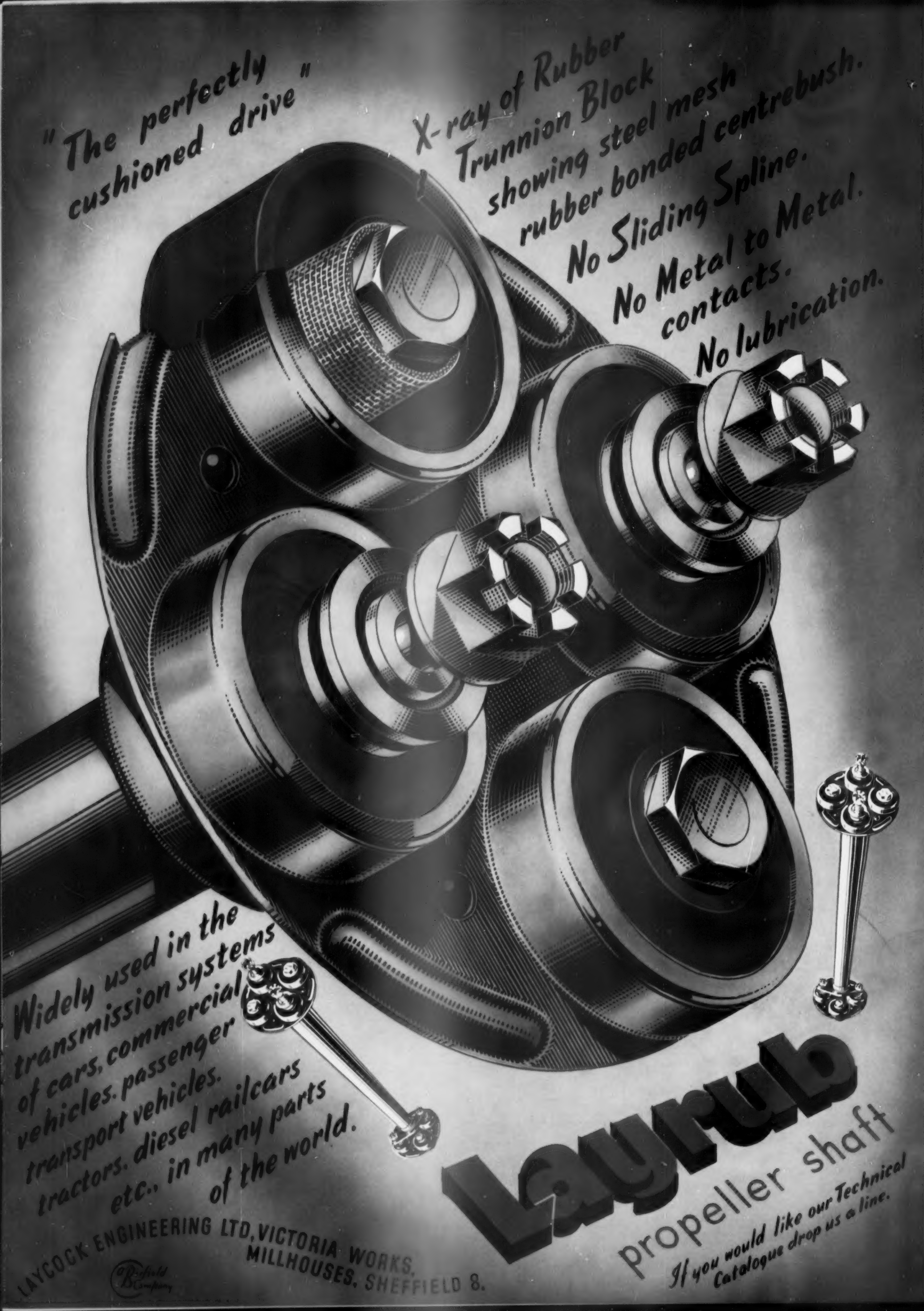

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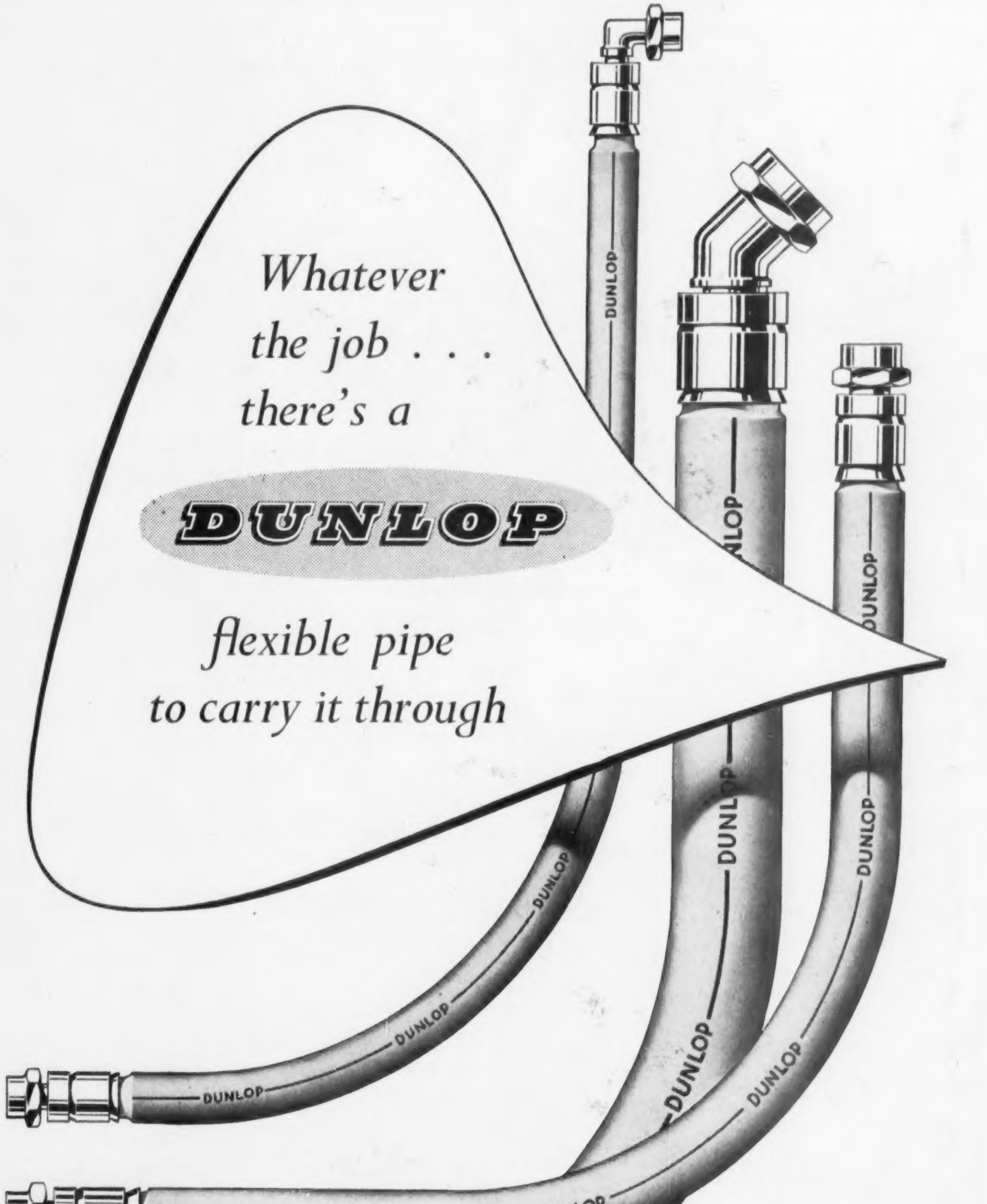

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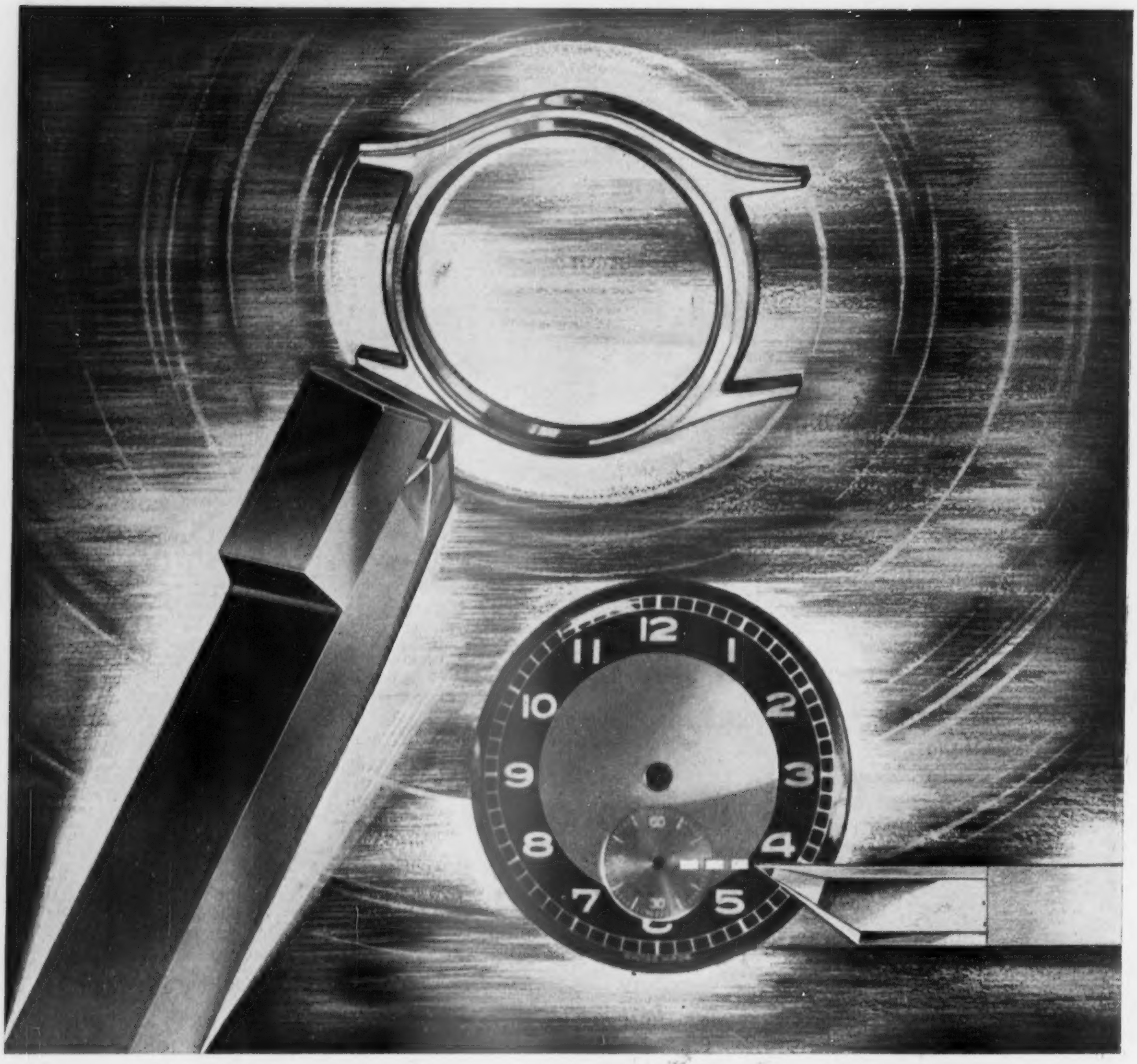

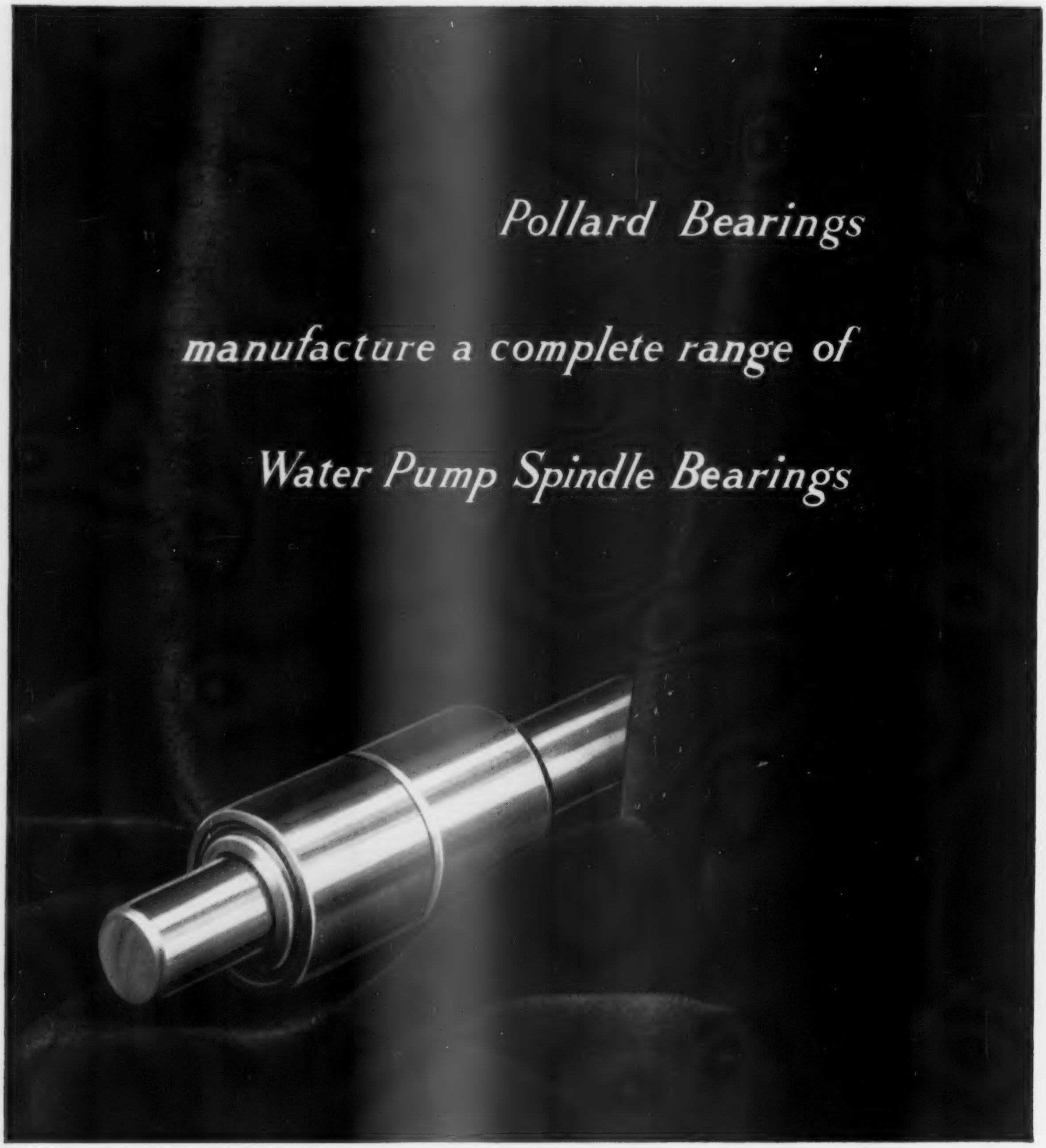
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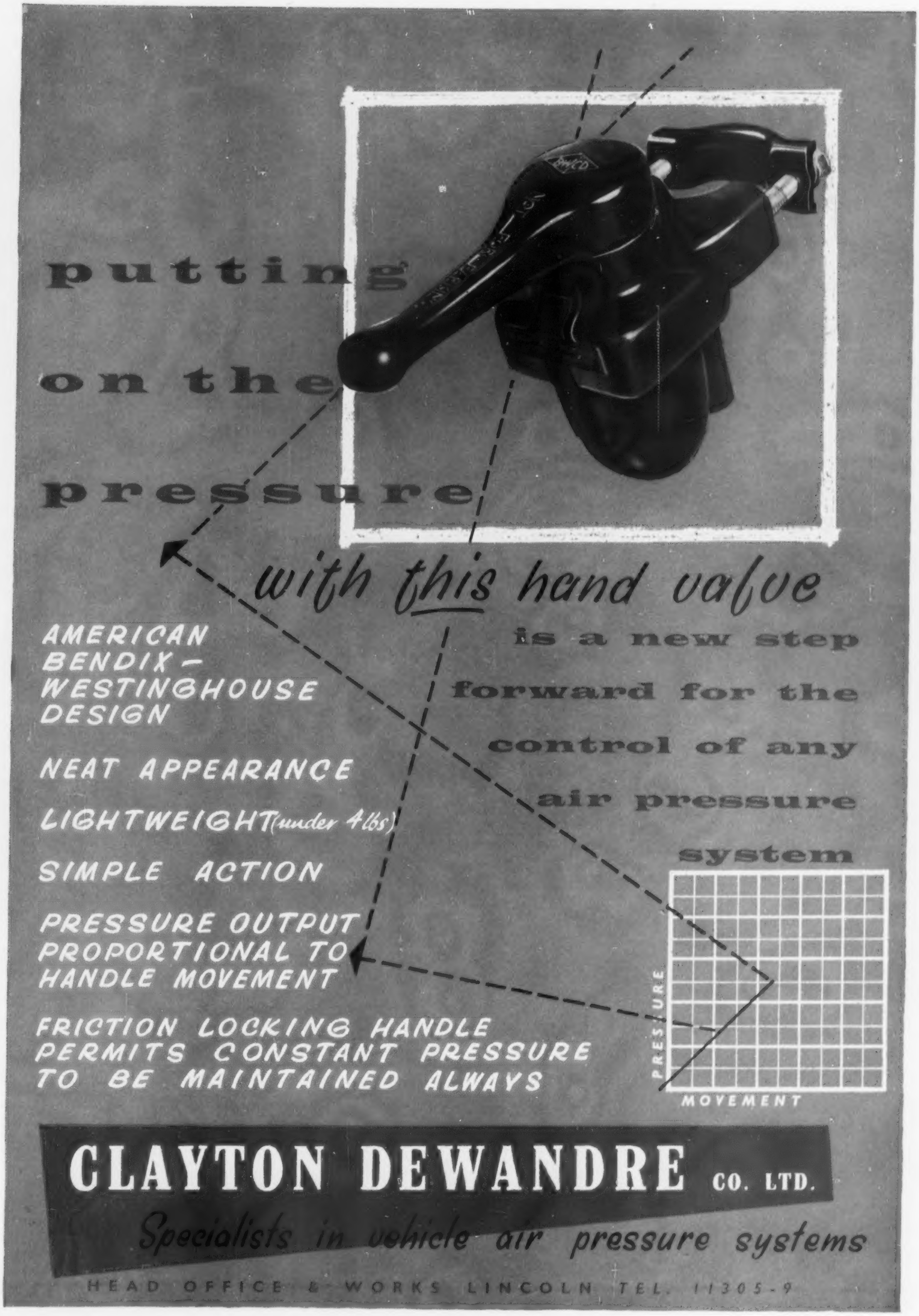
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AUTOMOBILE ENGINEER

Design, Materials, Production Methods, and Works Equipment

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Automatic Production Lines

HOW far the technique designated "automation" by the Americans will be adopted in this country is still a matter of conjecture. Certainly it will not develop so quickly nor be as widely adopted as in the U.S.A. There are two reasons for this, both matters of economics. In the first place, British manufacturers have not the same compelling reasons for reducing labour costs as have their American counterparts, since labour costs here represent a much smaller proportion of total costs. Secondly, the prospective output of any one British model is much less than the outputs for which automation is applied in the U.S.A. As a consequence amortization charges will be relatively high.

Nevertheless, it is almost inevitable that the larger producers in this country will adopt some of the ideas now being applied in America; therefore it is worth while studying what American experience has shown to be desirable. At the outset it must be stressed that the technique can be economical only if the components round which it is built are obtained from other than American sources. Preferably, the sources should be British since the closest co-operation between builder and user will be essential. Furthermore, the pound/dollar rate of exchange has virtually priced American machines out of the British market, and British equipment would cost not more than one-half of the price of American equipment.

Machine trends

E. E. Opel of the National Automatic Tool Co. of America suggests that the wider adoption of the technique will lead to:—

(1) Machine units will tend to become smaller and be connected by automation.
(2) Means will be provided for stock piling between machine units or for by-passing parts through parallel machine units.
(3) More safety controls will be used.
(4) Machine operators will be patrol men. Instead of moving parts from one machine to another, they will patrol the production line watching for signs of impending trouble. These men must be more capable than the old type machine operator.
(5) Maintenance will probably not be greatly increased.
(6) Inspection will be a serious problem. Automatic inspection will be used for :— depth of holes, size of holes, location of holes and oil leaks.
(7) Machines will automatically eject defective parts.
(8) Machines will be automatically lubricated.
(9) There will be many safety features, both to protect the machine and to signalize what the operator should do when trouble occurs.
(10) Meters will be fitted to machines to indicate when tool changes are necessary.

Reliability is probably the most important single factor. A single transfer machine may cost £50,000 or more, yet if a single control device fails, the entire machine stops and there is a complete cessation of production. Thus an item costing only two or three pounds may cause heavy losses through interrupted production and possible damage to the machine. Reliability is also necessary to reduce maintenance costs. If there is not a sufficient degree of reliability, the unskilled or semi-skilled operators released by automation will have to be replaced by highly trained service technicians.

Control panels

The component that at present is most subject to excessive maintenance is the control panel. Therefore great care must be given to its location to ensure that it is protected from dirt, chips, coolant and oil thrown off by the machines. The panel must be readily accessible at all times and be so mounted that the maintenance man has an unobstructed view of the machine while he is working on the panel. This is important because it may be necessary to run the machine, or parts of the machine, through a cycle of manual operation of contactors or other controls and the maintenance man must be able to watch the machine movements.

Experience has shown that the control panels should embody the following:—

(1) All possible control equipment for each head should be mounted together as for a single unit panel.
(2) When sectionalizing disconnect switches are used, they should be located behind the panel door for the control affected.
(3) On multi-door panels, control apparatus should not be mounted behind door frames or gutters where removal or service is difficult.
(4) Interlocks with doors and disconnect switches should be so arranged that the switch must be open before the door can be opened, but so arranged that the door can be latched closed when the disconnect switch is closed.
(5) To avoid contamination from dirt dropping from

moving and wearing parts, terminal strips should be mounted in a vertical plane at the side of their panel section and not horizontally.

(6) Lighting should be installed inside the panel enclosure to facilitate maintenance.

(7) Over-current protection should be provided for conductors to solenoids.

(8) All control equipment should be so spaced as to allow room and clearance for mounting the next size larger control device.

Minor Details

IT may seem to be stressing the obvious to say that every item of a new vehicle, even the most insignificant, should be given due consideration while the project is still in the design stage. Then, when the prototypes have been made only one or two features should require modification in the light of experience gained from development tests. Nevertheless, a study of models produced even by some of the largest manufacturers in different countries suggests that this methodical approach is not always applied as consistently as it ought to be.

Rear view mirrors, for example, are frequently neglected during the initial stages of design; they are added rather as an afterthought at the prototype stage. The result is that in some vehicles the mirror is so near the centre of the windscreen that it causes a large blind spot in an important area of the driver's field of vision. In other models the mirror may be clear of the windscreen, but give an inadequate range of rearward vision. Obviously the rear view mirror arrangement should be planned as soon as the stylists and body engineers in conjunction have settled the glass positions and dimensions. At that stage, it is easy to make minor modifications to the glass dimensions in the interests of adequate all-round vision.

For many years, certain manufacturers in this country have been making a careful study of the problems of visibility from the driving seat, but it is doubtful whether all could produce permanent records in the form of charts showing the range of visibility in all directions for the models they have developed during the last decade. Yet records of this type are most useful as standards against which to compare new designs.

Despite its importance, seating is often designed to fit the car rather than the human body. Then, if complaints are received from the development test department, it is often impossible to provide the optimum of comfort because of limitations imposed by the interior dimensions and arrangement of the vehicle. Elsewhere in this issue, there is an article discussing the structural characteristics of the human frame in relation to the driving seat. It is to be hoped that this will stimulate further investigation, since a careful and comprehensive study of the subject might make possible the formulation of general rules that would take much of the guesswork out of seat design.

Nor is it the body designer alone who must plan ahead for every detail. As most car owners know only too well, chassis components such as jacking points, grease nipples and filler plugs are often in relatively inaccessible positions. Because of current body styling these features need even more careful arrangement now than in the past.

Most new models show evidence that considerable thought has been given to accessibility. For example, contact breaker and distributor units are now almost invariably mounted on turret castings to bring them to more accessible positions, and in many vehicles the battery is mounted on a bracket near the front of the wing valence so that the acid level can be checked readily when the bonnet lid is open. Nevertheless, it is still possible to criticize adversely certain features of almost every new model, and the scope for such criticism can be reduced only by a methodical approach to the many problems.

From the engineering point of view, features of the types discussed must be regarded as minor details, but they impinge upon the car user much more than the important engineering details. The mechanical reliability of the modern car is more or less taken for granted; therefore the irritations caused by minor details assume a disproportionate importance.

CONTENTS

THE STANDARD EIGHT

Part I. Details of the 803 cm³ Engine

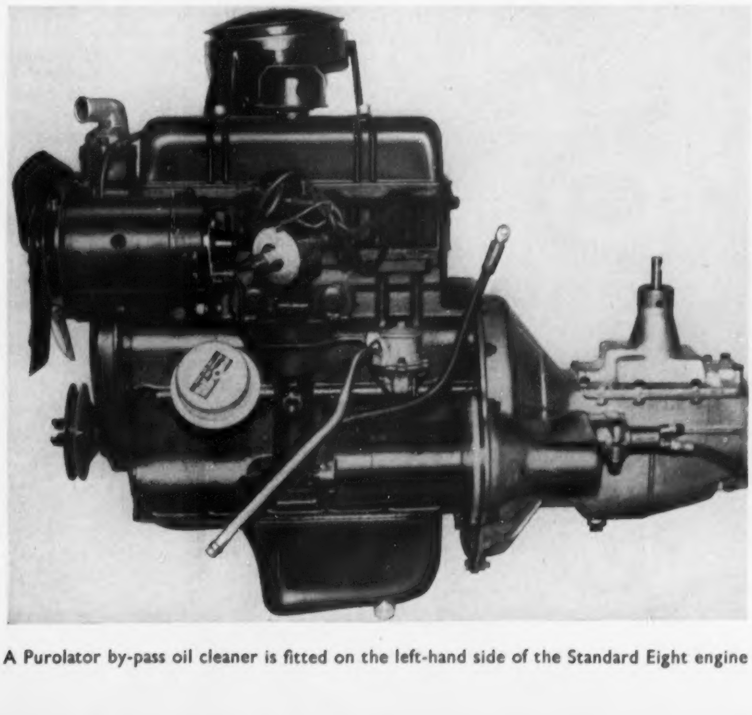
THE power unit for the new Standard Eight is an overhead valve engine with a swept volume of 803 cm^3. It was designed specifically for this chassis and is not, as so often is the case, an existing unit modified to suit a new vehicle. As a result, in designing this engine, the manufacturers have been able to take full advantage of the latest technical developments.

Contrary to views widely expressed elsewhere, square cylinder dimensions are not necessarily the ideal to be aimed at. Had these proportions been adopted in the Standard Eight, the length of the unit would have been increased. In the new engine, adequate space for the valve ports has been obtained by arranging the combustion chambers so that they extend longitudinally beyond the bounds imposed by the cylinder bores at the centre and ends of the block. This has been done without reducing the gasket width between the cylinders: in fact, the narrowest section of gasket is between the adjacent sides of the cylinders on which the combustion chambers do not overhang, that is, between numbers 1 and 2, and between numbers 3 and 4 cylinders. Since it is the exhaust ports that are overhung, there is no loss of volumetric efficiency. Moreover, the water jacket space beneath the overhung portions of the combustion chambers is adequate to ensure that local hot spots are not formed. As the inlet valve ports are within the projected area of the cylinder bores there is no obstruction to the inflowing gas.

SPECIFICATION

ENGINE: Four cylinders. Bore and stroke 58 mm × 76 mm. Swept volume 803 cm^3. Maximum b.h.p. 28 at 4,500 r.p.m. Maximum b.m.e.p. and torque respectively 120 lb/in^2 and 470 lb-in at 2,800 r.p.m. Compression ratio 7.0:1. Three bearing crankshaft, dynamically balanced. Overhead valves push rod operated. Solex downdraught carburettor with 18 mm choke. A.C. Delco diaphragm fuel pump.

From the data given in the specification panel, it can be seen that the stroke:bore ratio is 1·31:1 and the connecting rod length:stroke ratio 1·925:1. At the engine speed at which maximum b.h.p. is developed, the mean piston speed is 2,320 ft/min. The maximum b.m.e.p., of 120 lb/in^2, is developed at 2,800 r.p.m. In terms of b.h.p/in^2 piston area and b.h.p/litre, figures of 1·7 and 34·9 are obtained respectively. The engine dry weight is 248 lb, so 0·113 b.h.p. is developed per pound weight. A minimum brake specific fuel consumption of 0·59 pt/b.h.p-hr is obtained. With the air filter and flywheel fitted, the overall dimensions of the engine are: length $22\frac{1}{4}$ in, width $17\frac{1}{2}$ in, height $25\frac{1}{4}$ in.



A Purolator by-pass oil cleaner is fitted on the left-hand side of the Standard Eight engine

The power unit is installed with its longitudinal axis 6 deg from the horizontal. At the front, a conventional V-arrangement of metal-to-rubber bonded sandwich mountings is employed. The sandwich units are secured, by two bolts on each side of the engine, to extensions of the front plate. They are carried on brackets on the side members of the forward structure. At the rear, a slotted Metaxentric mounting is carried in an eye on top of the rear extension of the gearbox casing. A bolt is passed transversely through the mounting to secure it between lugs welded to the front cross member of the body-chassis structure.

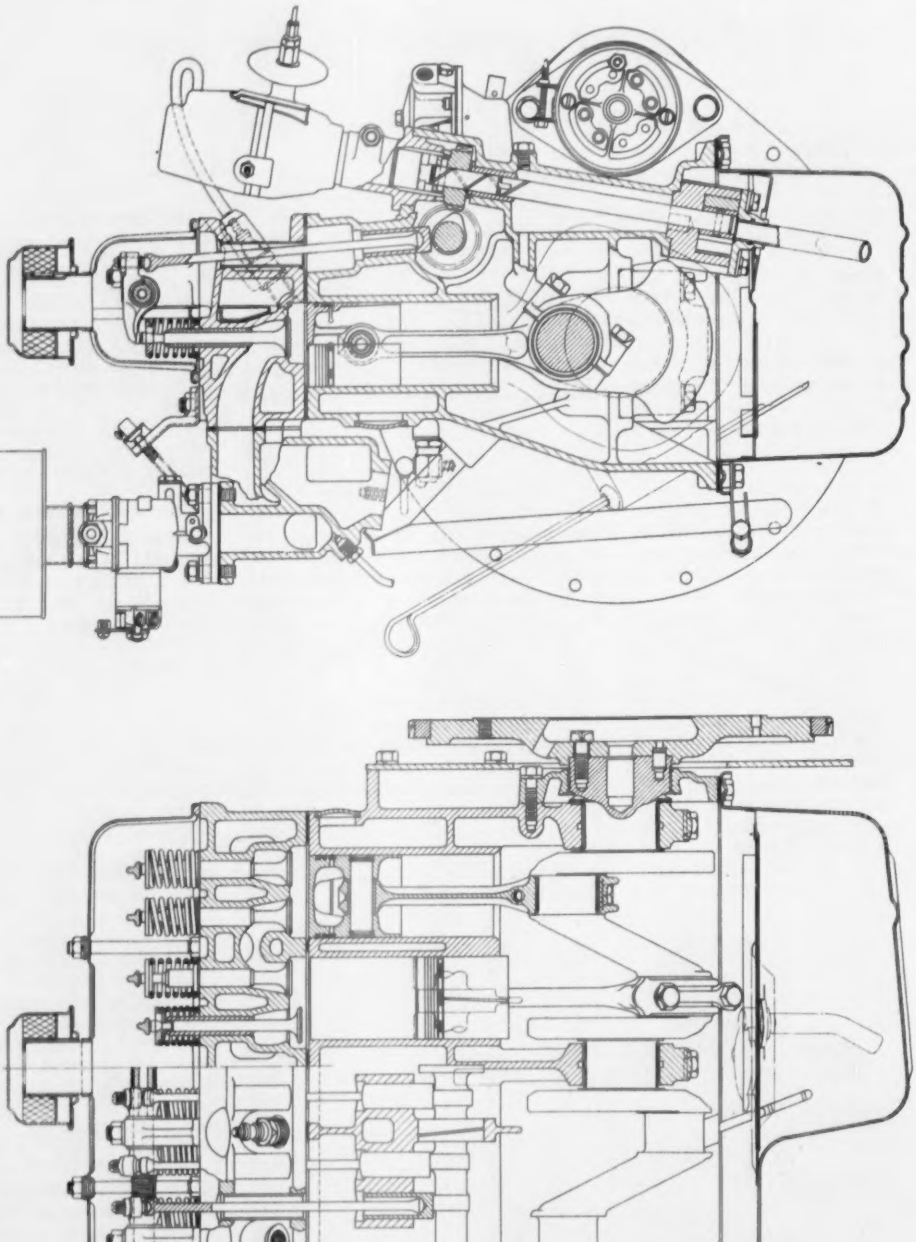
Cylinder block and crankcase

An integral cylinder block and crankcase unit of B.S. 1452, grade 14 cast iron is employed. The front face is machined to take the engine plate, which is assembled to it together with a 0·015 in thick Tan fibre washer and the timing drive cover. Another machined face at the rear takes the bellhousing with its closing plate, which is located by two $\frac{3}{8}$ in diameter dowels. The joint is sealed with a 0·006 in thick paper washer. Between the crankcase and the sump, a $\frac{1}{16}$ in thick Tan fibre washer is fitted. The cylinders are cast integrally with the block, and the minimum jacket space between them is $\frac{1}{4}$ in. Their walls are $\frac{3}{16}$ in thick and the bores are hone finished. The top decking is $\frac{5}{16}$ in thick.

A deep skirted crankcase design has been adopted, the sump joint face being $2\frac{3}{4}$ in below the axis of the crankshaft. The bearing caps are each secured by two $\frac{7}{16}$ in diameter, B.S. 1768R set bolts locked by spring washers. Lateral location is provided by their registering between $\frac{3}{8}$ in deep shoulders in the front and rear walls and intermediate web of the crankcase. At the front, a zinc alloy sealing block beneath the bearing cap closes the end of the crankcase. Its housing is coated with shellac on assembly, and its ends are grooved to receive hardwood sealing pieces. The rear seal is formed by a one-piece, bolted-on aluminium housing in which a hole is bored to accommodate the end of the crankshaft. Oil return scrolls are machined both in the bore in the housing and round the shaft.

Crankshaft, connecting rods and pistons

A forged, En 111, three-bearing crankshaft is employed. The journals and crank pins are hardened to 277-311 Brinell. Axial location is effected at the rear, where Vandervell V32 semi-circular thrust washers are fitted in recesses machined in the front and



GENERAL ARRANGEMENT OF THE STANDARD EIGHT ENGINE
Bore and stroke: 58 mm by 76 mm Swept volume: 803 cm³

rear faces of the bearing housing. The main journals are 2·0010-2·0005 in diameter and Vandervell J 3218 M shells are fitted. These shells are located in the usual manner by tabs pressed out at the abutting faces on one side. The tabs register in slots in the cap and housing. All three bearings are $1\frac{1}{8}$ in long and the diametral clearance is 0·001 in minimum.

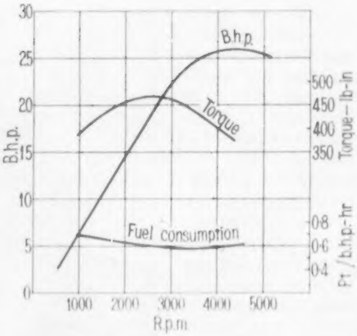
From the front of the front web to the back of the rear web, the overall length of the crankshaft is 11·77 in. The webs are $\frac{5}{8}$ in thick by $2\frac{1}{4}$ in wide measured across the journal axes, and $2\frac{1}{4}$ in wide measured across the crank pin axes. The inclined crank arms are 1 in by 2 in rectangular section. Each crank pin is 1·6255-1·6250 in diameter. An oil thrower lip and an oil return scroll are machined near the tail end of the shaft. An Oilite bush in a counterbore in the rear end carries the front of the gearbox primary shaft.

A B.S. 1452, grade 17 cast iron flywheel is spigoted on to the end of the shaft. It is secured by four $\frac{3}{8}$ in diameter set bolts locked by tab washers and is located by one $\frac{3}{8}$ in diameter dowel. The principal dimensions of the flywheel are: overall diameter $11\frac{13}{16}$ in over the ring gear, thickness adjacent to the ring gear $\frac{11}{16}$ in, weight $15\frac{3}{4}$ lb, mass moment of inertia 1·86 lb-ft^2. An En 8D starter ring gear is shrunk on and located against a shoulder on the rim. It has 117 teeth and meshes with a starter pinion with nine teeth.

The centre-to-centre length of the En 8R, H-section connecting rods is 5·75 in. Their cross sectional dimensions are: overall depth $\frac{13}{16}$ in, width over flanges $\frac{1}{2}$ in, web thickness approximately $\frac{1}{8}$ in, and the weight of each is 1·27 lb. The big ends are split at an angle of 38 deg from the axis of the rod, and the caps are located by hollow dowels round their $\frac{3}{8}$ in diameter, En 111 holding-down bolts which are secured by locking plates.

Vandervell D2, steel backed, bimetal bearing shells are fitted. They are $\frac{7}{8}$ in long and the minimum running clearance allowed is 0·0005 in. Clevite 10 bushes are pressed into the small ends. These bushes are 0·865-0·875 in long and the diametral clearance between them and the gudgeon pins is −0·00035 to +0·0003 in. Countersunk holes are drilled in the ends of the rods to pass splashed lubricating oil to the bushes. The En 32, case-hardened gudgeon pins are 0·367-0·359 in and 0·62510-0·62485 in inside and outside diameter respectively. In each piston boss the bearing length is $\frac{1}{2}$ in. The usual arrangement of Seeger type circlips in the bosses limits the axial float.

The pistons are supplied by the Automotive Engineering Ring Co. Ltd. and the British Piston Ring Co. Ltd. They are of the split-skirt type and have flat crowns, are made of Y-alloy and weigh 8 oz $5\frac{1}{2}$ drm. Two compression rings and one oil control ring are fitted above the gudgeon pin. The square faced compression rings are of 4K6 cast iron. Their gap when fitted



Engine performance curves taken with the air cleaner, fan, dynamo and water pump installed and using the test bench exhaust system

is 0·012-0·007 in and their face width is 2·000-1·975 mm (0·0787-0·0777 in). The radial thickness of these rings is 0·094-0·088 in, and the diameter of the piston at the base of the ring grooves is 2·078-2·074 in. A side clearance of 0·003-0·001 in is allowed. The dimensions of the oil control rings, which are made of D.T.D. 485, are the same as those for the compression rings except that the face width is 0·1563-0·1553 in and the side clearance is 0·0027-0·0007 in.

Timing gear, camshaft and valve gear

A pressed steel timing cover is employed. Formed in it is the housing for the lip type oil seal that bears on the boss of the B.S. 1452 grade 10 cast iron fan-belt pulley. The pulley is carried, together with the En 8R timing drive sprocket, on a 1 in diameter extension of the crankshaft. Interposed between the bosses of these two wheels is a dished thrower ring; both are driven by a single Woodruff key. The whole assembly is pulled against the shoulder of the front main journal bearing by a special nut on the $\frac{3}{4}$ in diameter threaded front end of the shaft. This nut incorporates the dogs for the starter handle.

The drive is transmitted to the half speed wheel by a $\frac{3}{8}$ in pitch chain, supplied by the Renold and Coventry Chain Co. Ltd. The half speed wheel is a flat disc spigoted on to the end of the camshaft and secured by two $\frac{5}{16}$ in diameter set bolts. There are four holes for these bolts and they are arranged so that an adjustment equivalent to $\frac{1}{4}$ of a tooth can be made to the position of the wheel relative to the shaft. As there are 42 teeth on the wheel this adjustment represents 2·14 deg.

A Monikrom cast iron camshaft is fitted. Axial location is effected by a semi-circular, 5 S.W.G., En 2 plate that registers in a groove round the front end of the shaft. Two bolts secure this plate to the crankcase. All four camshaft journals run directly in bosses in the block which are bored 1·8448-1·8433 in to receive them. The front bearing is 1·13 in long, the two intermediate ones are 0·63 in long and the length of the rear one is 0·88 in. In each, the running clearance is 0·0046-0·0026 in. Between the bearings, the camshaft diameter is $\frac{7}{8}$ in.

The cam characteristics at 6,440 r.p.m., which is the valve crash speed, are as follows: maximum positive acceleration of tappet (flank) 12,930 ft/sec^2, maximum negative acceleration of tappet (nose) 4,530 ft/sec^2, lift of cam 0·170 in, nominal period of cam 120 deg. The drive gear for the oil pump and contact breaker and distributor is machined on the shaft between the two intermediate bearings.

A one-piece inlet and exhaust manifold is fitted on the right-hand side of the engine

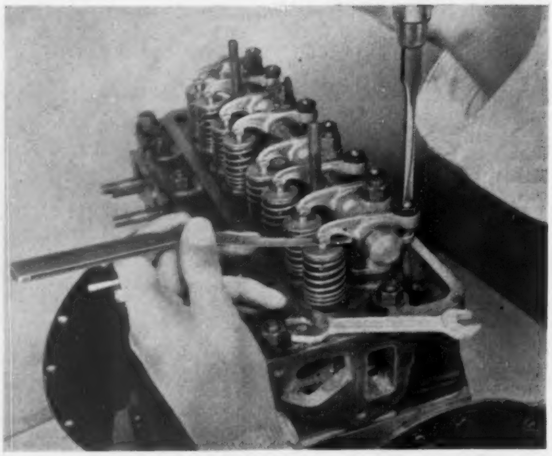
It has 14 teeth, a helix angle of 46 deg, and a pressure angle of 20 deg (normal). Its backlash is 0·006 in and normal diametral pitch 12/14.

Piston type tappets of chilled cast iron are employed. Their housings are 0·6880-0·6873 in diameter, and the diametral clearance in each is 0·0013-0·0002 in. The $\frac{5}{32}$ in diameter by $7\frac{3}{4}$ in long, En 8R push rods have spherical lower ends where they seat in the tappets, and their upper ends are of cupped form to receive the spherical ended En 32B tappet adjusting screws.

Four pedestals, of Birmal P83 aluminium alloy, carry the rocker shaft assembly. Aluminium alloy is employed to minimize variations in tappet clearance with engine temperature. As the valve stems become hot and expand, so also do the pedestals which, although they are at a lower temperature, tend to expand approximately the same amount because of the high coefficient of expansion of the material of which they are made. Each pedestal is held down by a single, $\frac{3}{8}$ in diameter stud; this is adequate to clamp the shaft in the bore. Had iron, instead of aluminium alloy, been used for the pedestals, the bosses would have had to have been split to obtain this clamping action.

The rockers are of En 32B. They bear directly on the shaft, which is also of En 32B. Except in their bores and on the end pads, they are copper plated and then carburized. The copper plating localizes the carburizing treatment. This shaft is also carburized and hardened locally at the bearing surfaces for the rockers, and flats are machined on it to spread the oil over the rocker bores. Between the shaft and the rocker bores the diametral clearance is 0·0023-0·0008 in, and the shaft inside and outside diameters are respectively $\frac{1}{4}$ in and 0·5612-0·5607 in.

Caps, machined from $\frac{7}{8}$ in diameter En 3B bar, are placed over the ends of the rocker shaft and secured by split pins. They serve the dual purpose of sealing the ends of the hollow shaft and retaining the components on it. On assembly, first one end-cap is placed in position, followed by a double spring washer and an end pedestal. Next, all the rockers and pedestals, together with the compression coil springs between the rockers, are assembled on to the shaft; finally another double spring washer and end cap are put on to complete the assembly, which is then ready for mounting on the cylinder head. The shaft is located by a countersunk screw fitted, together with a conical Shakeproof washer, in the rear pedestal. A countersink for this screw is cast in each pedestal but only the rear one is drilled, and the screw is tightened into a tapped hole in the shaft.



Tappet adjustment, if carried out methodically, is a simple operation

Cylinder head, manifolds and carburettor

A B.S. 1452 grade 14 cast iron cylinder head is employed. The overall depth of the casting is $3\frac{1}{16}$ in and its overall width is $5\frac{7}{8}$ in. Eleven, $\frac{3}{8}$ in diameter, En 111T, holding down studs are fitted; they pull the head down on a copper and asbestos gasket. The pressed steel rocker cover is secured by two studs screwed into the cylinder head. It is tightened down on to a Neoprene, synthetic rubber and granulated cork joint washer, but flexible washers are not fitted between the holding down nut and the cover.

Combustion chambers of the type commonly known as "inverted bath tub" are incorporated in the head. They are offset towards the left-hand side so that an inverted shelf is formed on the right-hand side above the piston. This gives a squish effect. The valves are carried vertically in line. Between the adjacent inlet and exhaust valve stem axes the distance is $1\frac{5}{16}$ in. The springs are retained by Holdsworth washers. Other details of the valves are given in the accompanying table.

Siamesed inlet and separate exhaust ports are incorporated and all are completely encircled by the water jacket. A distribution tube is carried in the head to direct coolant on to the exhaust ports and spark plug bosses. The push rods, where they pass through the head, are housed in vertical tubes, fitted between flange extensions at the top and bottom of the left-hand side of the head. The ends of these tubes are belled into countersunk holes in the flanges.

The inlet and exhaust manifolds are cast integrally and are made of B.S.1452 grade 10 cast iron. A Cemjo, steel-faced asbestos joint washer is fitted between the manifolds and the head. A hot spot is formed at the junction between the base of the inlet riser and the exhaust manifold. The exhaust pipe is $1\frac{1}{8}$ in diameter.

A somewhat unusual crankcase breathing system has been adopted. In the oil filler cap, which is on top of the rocker cover, is a filter element; a breather tube from the crankcase extends approximately to the level of the base of the sump. A forward facing air intake on the filler cap ensures that, while the engine is running, the fan blows air in through the filter and it passes out through the breather pipe. Furthermore, it is said that since the resistance offered by the filter element to the entry of air is greater than that of the breather tube, the tendency for condensation to take place in the rocker box when the engine has stopped is minimized.

A Solex, 26ZAICO downdraught carburettor is mounted on a 0·19 in

The lip on the wall of the crankcase, to the right of the pump in this illustration, prevents the discharge of oil from the by-pass filter on to the crankshaft

thick, hard bonded-asbestos washer between two 0·015 in thick Permanite Aero washers. The choke diameter is 18 mm and the carburettor jet settings are as follows: main 100, correction 200, auxiliary 40, auxiliary air bleeds 1·0 GA8 (2) and GS110, starter bleed 3·0. A 1·5 mm needle seat is fitted. The fuel is drawn from a 7 gallon tank by an A.C. Delco, diaphragm-type pump. An A.C. Delco E/AC5659 air cleaner is mounted on top of the carburettor.

The throttle control system is of interest in that it is completely interchangeable for left- and right-hand drive vehicles. For the left-hand drive arrangement, the end of the cable is attached to the lever on the butterfly control and the casing to a bracket on the carburettor, while for the right-hand drive, the casing is attached to the control and the cable to the bracket. This arrangement, of course, not only simplifies the assembly process but also represents a small saving in tooling costs. Moreover, it makes easier the servicing spares supply problem.

Water pump and cooling system

A rubber V-belt, reinforced with nylon cord, transmits the drive to the B.S. 1452 grade 10 cast iron pulleys on the water pump and dynamo. It is $\frac{3}{8}$ in wide by $\frac{5}{16}$ in thick and the V-angle is 40 deg. The pump is driven at 1·25 times engine speed and is housed in a B.S. 1452 grade 10 cast iron body bolted to the front of the cylinder head. A water outlet and thermostat housing is also incorporated in the pump casting.

The $\frac{5}{8}$ in diameter En 111T pump spindle is carried in two sealed ball bearings, the centres of which are spaced about $1\frac{1}{16}$ in apart. The assembly is inserted from the front into the nose piece of the pump body casting. The outer race of the rear bearing is located against a shoulder in its housing and that of the front bearing is retained by a circlip in a groove just inside the bore of the housing. The inner races are separated by a tubular distance piece. A grease nipple is fitted on top of the casting; it is screwed into a hole through which the lubricant is passed to the space between the two bearings.

Keyed on to the front end of the spindle is the pulley, to which is bolted the $12\frac{1}{2}$ in diameter, four-bladed, pressed steel fan. An unusual feature of the fan is that it is not spigoted on, but location is effected simply by the fit of the bolts in their holes. A self-locking nut on the $\frac{5}{16}$ in diameter end of the spindle pulls the whole assembly, comprising the pulley, bearings and distance tube, against a washer and circlip in a groove round the spindle.

Immediately behind the circlip is a rubber thrower ring which works in a drainage space in the casting. To the rear of that is the front wall of the impeller chamber, through which the spindle extends to carry the pressed-on B.S. 218 brass impeller. The impeller diameter is $2\frac{3}{4}$ in. Interposed between the front of the rotor and the front wall of the casing is a conventional, spring-loaded seal supplied by Super Oil Seals and Gaskets Ltd. The seal is carried in a cupped recess in the boss of the impeller; moulded into its front end is a carbon thrust-ring that bears on a ground face on the front wall of the impeller casing.

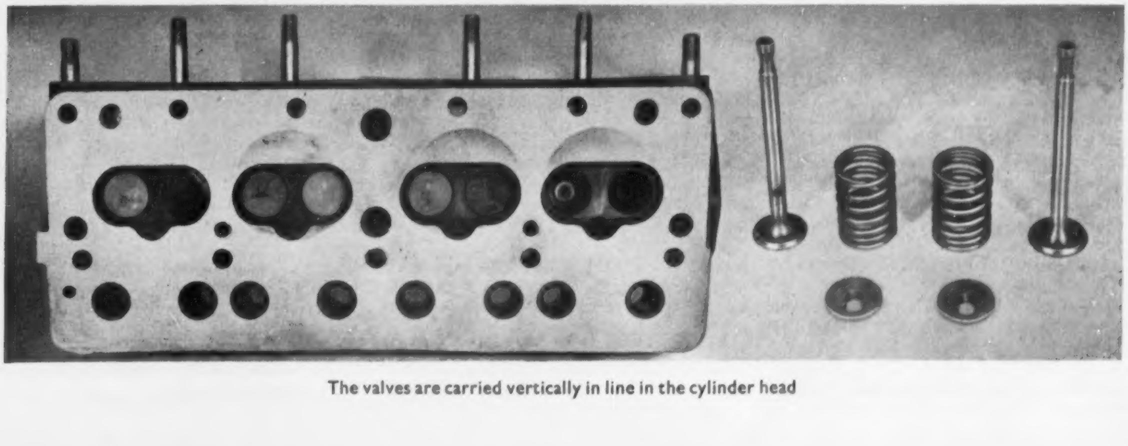
A pressurized cooling system has been adopted, and the pressure is maintained at $3\frac{1}{4}$-$4\frac{1}{4}$ lb/in^2 by an A.C. Delco, combined filler cap and blow-off valve. The radiator is of the finned tube type. Its frontal area is 254 in^2 and the block thickness is $1\frac{1}{4}$ in. Two rows of tubes are fitted. The capacity of the system is 7 pints, and the coolant, after passing from the radiator to the pump, is delivered to a distribution tube in the cylinder head. From there, the tube directs the flow round the exhaust valve seats and the spark plug bosses. Ducts between the head and the block are provided so that a subsidiary thermosyphon effect is set up to cool the cylinder walls. The coolant comes out past the thermostat valve, which lifts at 170-179 deg F, and thence back to the radiator.

VALVE DATA

	Inlet	Exhaust
Material	Sil. Chr. steel En 52	XB Valve steel En 59
Head diameter	1·121 –1·117 in	0·996 –0·992 in
Throat diameter	1·005 –1·000 in	0·880 –0·875 in
Stem diameter	0·3110–0·3100 in	0·3090–0·3080 in
Diametral clearance guide in	0·003 –0·001 in	0·005 –0·003 in
Seat angle	45 deg	
Valve seat material	cut directly in head	
Spring material	En 49D, D.T.D.5A	
Spring rate	148 lb/in	
Spring length, free	1·62 in approx	
Spring length, installed	1·375 in	
Spring surge frequency, fitted	33,000 osc/min	
Number of coils	4 working, $1\frac{3}{4}$ damper and 2 end coils	
Coil inside diameter	$0{\cdot}795 \pm 0{\cdot}010$ in	
Wire gauge	0·135 in	
Valve lift	0·255 in	
Rocker ratio	1·5 : 1	
Valve crash speed	6,440 r.p.m.	
Valve guide material	B.S. 1452 grade 14	
Valve guide length	2·44 in	
Valve guide inside and outside diameters	0·313–0·312 in and 0·502–0·501 in	
Tappet clearance	0·010 in cold	
Valve opens	10 deg B.T.D.C.	50 deg B.B.D.C.
Valve closes	50 deg A.B.D.C.	10 deg A.T.D.C.
Ignition timing	10 deg B.T.D.C., engine at rest	

Oil pump and lubrication system

A Hobourn Eaton pump is fitted, with its base approximately $\frac{1}{2}$ in above the oil level. Its cast iron body is spigoted and bolted to a machined face on the base of the intermediate web of the crankcase. The cast iron rotors are 1 in long. Pegged into the inner rotor, which is 1·598-1·599 in diameter, is the 0·499-0·498 in diameter En 8 driving spindle. Not only does the peg transmit the drive, but it also prevents the spindle from being forced upwards by oil pressure. A cast iron end-plate is bolted up to the base of the pump. It carries a $\frac{5}{16}$ in diameter suction tube, the upper end of which is belled into a countersink in the casting. A synthetic



The valves are carried vertically in line in the cylinder head

rubber sealing ring is fitted in an annular groove round the bore in which the tube is housed.

The driving spindle extends up through a chamber cored in the crankcase. Its slotted upper end projects about $\frac{7}{16}$ in into a phosphor bronze bush in a boss at the top of this chamber, and the lower end bears directly, for a length of $\frac{7}{8}$ in, in the pump casting. A short, En 32B spindle bears in the bush for a length of $\frac{3}{4}$ in, and a tongue on its lower end registers in the slot in the top of the drive spindle. The cast iron spiral gear that meshes with the drive gear machined on the camshaft is carried on this short spindle, to which it is secured by a $\frac{3}{16}$ in diameter peg. With this arrangement the gear and spindle assembly can be inserted from above, independently of the pump and its drive spindle which, of course, are assembled from below.

Axial thrust from the gear is taken by a flange at the upper end of the phosphor bronze bush, and the gear is held down by the lower end of the cast iron turret adaptor for the contact breaker and distributor unit. Slots in the end of this adaptor allow lubricant to pass out. The top of the gear boss is slotted to receive the offset-tongued sleeve round the spindle of the contact breaker and distributor unit. Offset tongues are employed so that the unit cannot be fitted with its timing 180 deg out of phase. The end of the spindle is spigoted into the gear boss.

A pressed steel sump of about 7 pints capacity is employed. Spot welded in it, about $\frac{3}{8}$ in above the oil level, is a pressed steel baffle, incorporating a 20 mesh gauze strainer of approximately 32 in^2 area. Lubricant is drawn through the inlet pipe to the pump and thence into the chamber round the drive spindle. A spiral groove round the spindle prevents an excessive amount of oil from passing through the upper bearing bush. The relief valve is screwed into a boss at the junction between the front of the chamber and the crankcase. It incorporates a low rate spring so that the provision of a means of adjustment has been unnecessary. A pressure gauge connection is fitted near the top of the chamber. The Purolator by-pass type oil cleaner is screwed into the gallery. Its outlet to the sump is shrouded by an overhanging lip formed on the crankcase wall to prevent the discharge from being directed on to the crank assembly.

From the chamber, the lubricant is passed to a $\frac{7}{16}$ in diameter main gallery and thence through 0·272 in diameter ducts to the main journals. The crankshaft is drilled $\frac{3}{16}$ in diameter from the main journals to the big ends. Small end lubrication is by splash. The ducts to the three camshaft bearings are 0·252 in diameter, but are reduced to $\frac{3}{32}$ in diameter adjacent to the bearings.

Each bearing has a spiral groove machined on it to spread the oil from a flat, which joins the ends of the groove, over the bearing surface. The rear bearing has two flats spaced so that they simultaneously uncover the oil inlet and outlet to the rocker gear, which thus is served by an intermittent feed from one flat, round the spiral groove to the other flat, and thence up through holes in the block and head to the rear pedestal. A hole drilled radially in the hollow, rocker shaft communicates with that in the pedestal to pass the lubricant into the shaft and out through more radial holes to the rocker bores.

HIGH SPEED PHOTOGRAPHY

A Useful Tool for the Analysis of Motion

HIGH speed motion-picture cameras are now being widely used in the American automobile industry for studying and analysing high speed motions. General Motors were the first automobile manufacturers to adopt this technique, but their example was soon followed by the other major companies.

The cameras, commercially available, are of robust construction and are easily portable. They are constantly being improved or modified for specific applications, and they are sold at less than many laboratory instruments of much more limited application. There is also a considerable amount of auxiliary equipment that has been developed to increase the adaptability of these cameras. The rates at which the pictures are taken are easily adjustable, since series motors are used and the voltage supply governs the picture-taking rate, whether the supply be direct or alternating current. Rates vary from 150 to 16,000 per second in frame sizes from 8 mm to 35 mm full-frame. The photographic quality of the rotating-prism cameras is approaching that of intermittent cameras in sharpness, contrast and resolution. In some high speed cameras, the steadiness and lack of weave, or bounce, is better than in intermittent cameras.

Units have been designed to control camera running time and the voltage supplied to the camera as well as to synchronize the event being studied with the camera. Improved film is being produced so that with the higher inertia films, grain size is kept to a minimum. Light sources, mainly tungsten-type incandescent, are also being constantly improved to allow both colour and black and white pictures to be made at smaller stops so as to gain the advantage of increased depth of field. In addition, the electronic flash has been synchronized to work with these cameras.

High speed photography has proved to be of great value in the study of spring behaviour. Although a spring may be designed to a standard formula, in service it might break after only a limited number of operations. It might have undesirable nodes and wave actions, or be maladjusted so that excessive vibration or bouncing might occur in parts connected with it, such as cams or followers. Before the use of high speed photography for scientific analysis, the normal procedure was to change the spring material, or to alter the thickness, or to make other changes that might seem desirable. This procedure was nothing more nor less than trial and error, and a great deal of time could be spent before the correct solution was found.

The technique is particularly suitable for studying the behaviour of coil springs, since the high speed camera makes it easy to study phenomena such as bounce of adjoining parts when the spring is operated. The spring tension may be a "hairline" adjustment with the result that the bounce does not appear at every operation—a condition that could not be detected by stroboscopic examination. A clutch spring might be incorrectly adjusted with the result that the engagement is erratic to the detriment of the clutch teeth or pads, and there may be secondary vibrations in the spring movement that cause it to chatter or fracture.

When a coil spring is linked with another unit, such as a valve lifter and camshaft, the problems become more involved and it is necessary to obtain measurements. If the valve spring is maladjusted, the valves will bounce and cause fuel to be fed at the wrong time, or an improper mixture of air and fuel to be taken into the cylinders. It may also cause the followers on the camshaft to bounce, and if this occurs regularly the camshaft will eventually be deformed. High speed photography is an excellent means for checking these conditions.

When this technique is employed, it should be remembered that it is usually advisable for each component part to be photographed separately, because it is extremely difficult, if not impossible, to comprehend fully the meaning of a large number of items moving simultaneously on a screen. With the improved light sources that are now available, colour film can be used as easily as black and white. Special printing or finishing to increase subject contrast is no longer necessary. The natural colours will themselves delineate the component parts.

MACHINING ENGINE DETAILS

The Production of Connecting Rods and Pistons for the New Ford Anglia and Prefect

AN appraisal of the manufacturing methods employed by Ford Motor Co. Ltd. at their Dagenham Works for the production of the new engine for Anglia and Prefect cars shows a continuance of the Company's general policy of maintaining a very high standard of quality at the lowest possible cost. For example, in the machine lines to be discussed in these notes it is found that considerable use has been made of existing machines, where such machines were known to be capable of handling the desired output. For those operations for which there were no machines already available, the best machines for the purpose have been obtained. Finally, for certain operations it was impossible to obtain standard machines that would meet the requirements as regards quantity and economy of output; for these, special Ford machines have been developed.

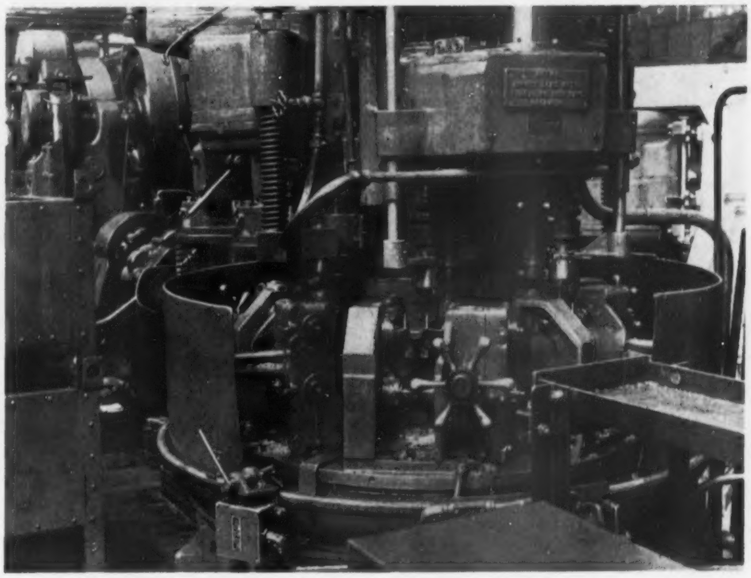


Fig. 1. Machining joint faces and studs on Ford connecting rods for Anglia and Prefect engine

Connecting rods

In contradistinction to the connecting rods for the Consul and Zephyr engines which are produced from a forging that comprises rod and cap, those for the Anglia and Prefect are produced from separate forgings for rod and cap. Furthermore, the rod has integral studs. Another point of interest is that the connecting rod assembly is of a type that, to the best of our knowledge, is not produced anywhere else in this country. Instead of having a thin wall bearing fitted in the big end, as is now normal practice, the babbit is deposited direct on the rod. This direct deposition of babbit on the rod was, of course, common practice at one time, but there was a very thick layer, whereas on these rods the babbit thickness is maintained within the limits 0·006-0·012 in.

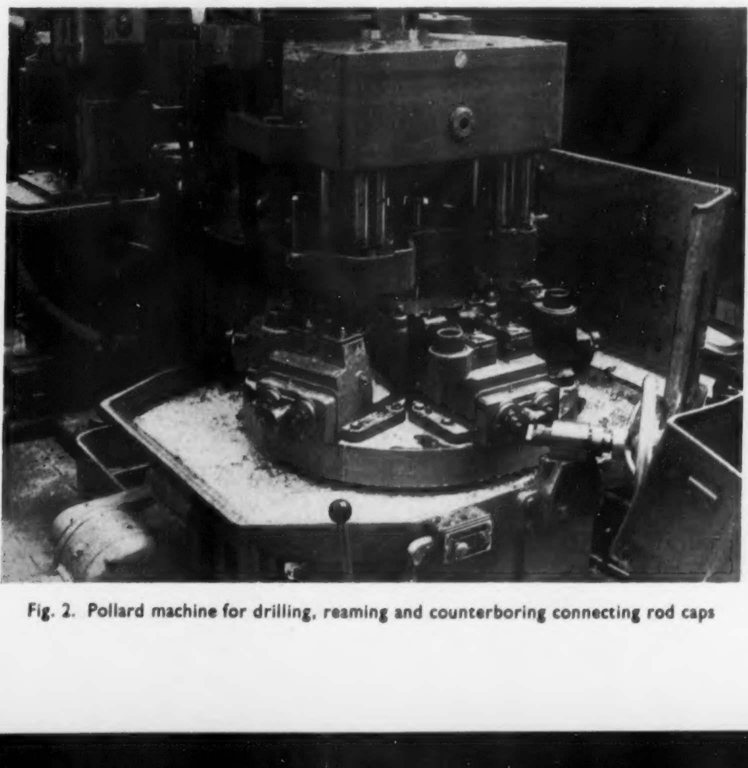


Fig. 2. Pollard machine for drilling, reaming and counterboring connecting rod caps

Before any machining is carried out on the rod, the forging is heated to 1500 deg F, quenched in oil and tempered at approximately 1000 deg F to give a Brinell of 269-302. It is then pickled, following which there is 100 per cent inspection for weight and Brinell hardness. Before the actual machining starts, there is a series of press operations at which both bosses and the studs are coined, the rod is straightened and both sides of the big end boss are sheared for locating centres.

The first machining operation is carried out on a Ford special centring machine tooled to drill simultaneously two centres in the sides of the small end boss and two in the sides of the big end boss. These centres are used as locating points for almost all the subsequent operations. It is important, therefore, that these centres are accurate both as regards position and dimensions. An interesting point in regard to these centres is that those in the large end are offset 0·062 in in relation to each other. This is to ensure that at the subsequent operations the rod can be loaded into the fixture only in one way, with the result that if any faults develop in machining, the cause can be traced much more quickly than would be possible if the rods could be loaded either way into the work fixtures.

A Greenlee multi-station machine is used for the first major operation on the rod. It is shown in Fig. 1. The work is loaded vertically in the fixture with the big end up. Each fixture has two fixed and two retractable centres. A single star wheel operates the retractable centres through rack and pinion motions. To prevent any possibility that the rod may be distorted through the application of excessive clamping pressure, the clamping mechanism incorporates a slipping device that becomes operative as soon as the specified clamping pressure is reached. This machine has six stations, one loading and five unloading. It is tooled

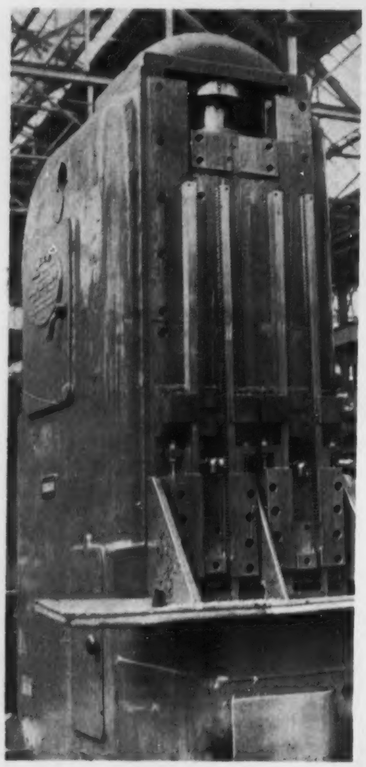


Fig. 3. Surface broaching machine for connecting rod caps

to rough and finish turn and chamfer the studs, to rough and finish face the joint faces and to thread the studs. On leaving the Greenlee machine, the ends of the studs are snagged on a pedestal grinder to remove burrs. The rod is then ready for assembly with the cap.

Before any machining is carried out on the cap, the forging is normalized at 1500 deg F and then air-cooled to give a Brinell hardness in the range 137-196. The machining operations are relatively simple and call for little comment. At the first, the joint faces are machined by means of a cutter mounted on a Pollard drill. A Pollard multi-head machine is used for the next operation. This machine, shown in Fig. 2, has five stations, four working and one for loading and unloading.

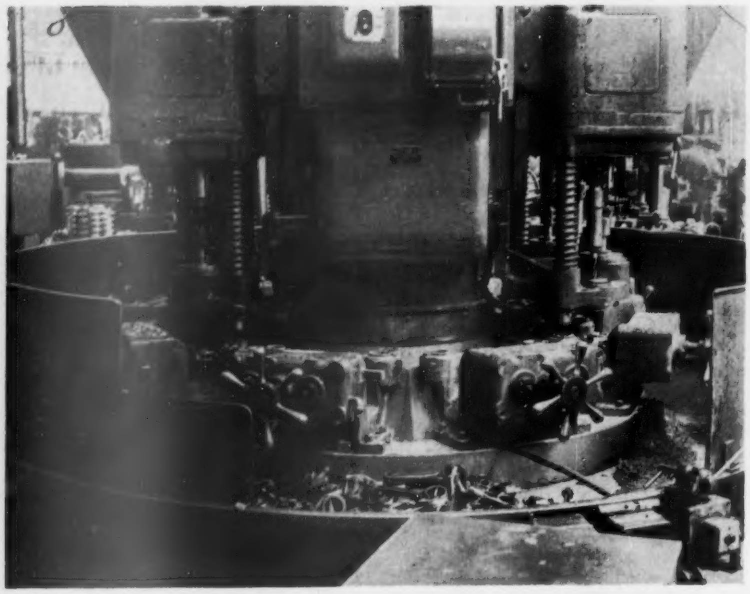


Fig. 4. Set-up for first boring operations on assembled rods

There are two components at each station, with location taken from the joint face and the O.D. of the bolt bosses. The operation sequence on this machine is: first station, load two components; second station, drill two holes in each component; third station, spot face the bolt bosses; fourth station, ream two holes in each component; fifth station, counterbore the holes in each component. Of the other operations on the cap as a separate component, it is only necessary to mention the manner in which the side faces are broached. A vertical surface broaching machine manufactured by Coventry Gauge and Tool Co. Ltd. is used. The interesting feature of this is that the work is not clamped down. Instead, it is merely placed on the work table with location taken from the joint face and from two dowel pins that register in the reamed holes. The cutting forces suffice to hold the work securely down on the locating face. This broaching machine, which is shown in Fig. 3, is a single slide machine, but it carries two sets of broaches so that two caps are machined simultaneously.

At this stage, the cap is assembled to the rod with paper shims between the joint faces. The nuts are tightened to 20-25 ft-lb by means of Desoutter torque units. The assembly is then transferred to a Greenlee multi-station machine, see Fig. 4, that is tooled for boring the large end and reaming the small end. Two components are

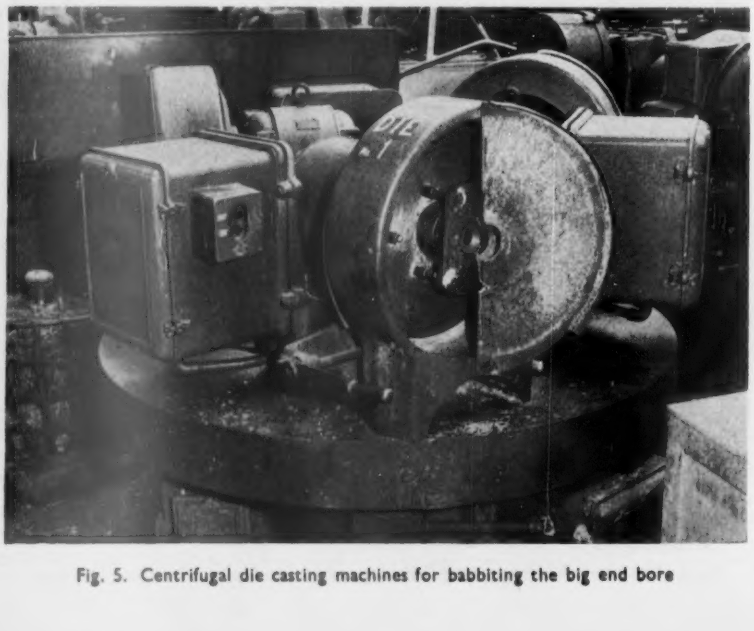


Fig. 5. Centrifugal die casting machines for babbiting the big end bore

mounted at each station to be machined simultaneously. Location is taken from the centres, and except that the rods are mounted horizontally in the work fixtures instead of vertically, the work-holding and clamping arrangements are similar to those used on the Greenlee machine referred to earlier.

Two further operations are carried out to prepare the rod for the deposition of the babbit. At the first, the large end is bored out on a Heald double-end boring machine. This machine is arranged to take four rods, all located from the centres. Two rods are machined simultaneously, and during the machining cycle, the other two work fixtures are loaded and unloaded. The second operation is carried out on a Hey double-ended drilling machine with cutters for facing both sides of the big end boss.

Before the babbiting sequence can be carried out, it is necessary to remove all oil and grease from the work. This cleaning is carried out in a trichlorethylene bath. After cleaning, the rod is pre-heated in a furnace and is then transferred to the tinning bath which is maintained at 600 deg F. From the tinning bath, the rod is immediately transferred to a Ryder three-station centrifugal die-casting machine, see Fig. 5. The babbit bath is maintained at 800 deg F, and is immediately adjacent to the pouring station.

As the centres are to be used as locations at subsequent operations, those in the big end boss are cleaned out immediately after the babbiting operation. Then, to facilitate the insertion of the small end bush, the small end bore is chamfered on one side. The bush is inserted on a Rhodes press. This press is also used to push a burnishing broach through the inserted bush. The sequence is that while the press ram is forcing a bush into the small end of one rod it is simultaneously forcing the burnishing broach through a bushed rod.

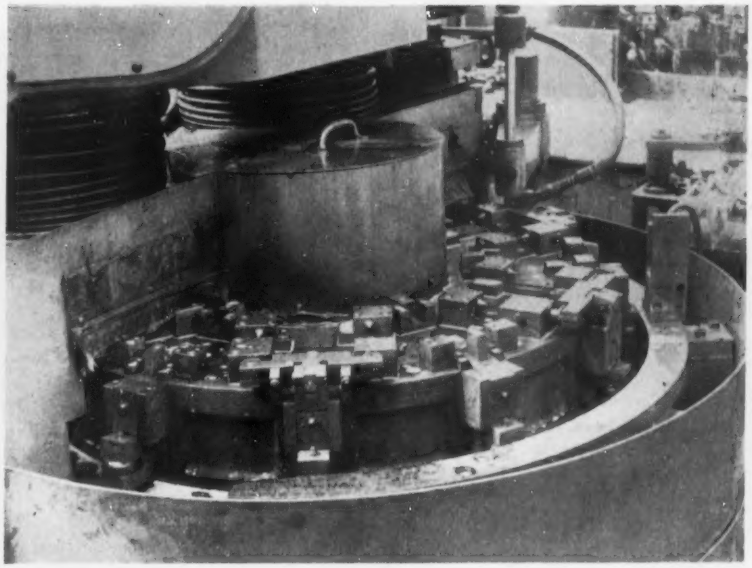


Fig. 6. Loading station for surface grinding side faces of connecting rod

A series of conventional operations are next carried out on the small end, in which both sides of the boss are faced, the oil holes are drilled and countersunk and the burrs are removed from the small end bore. The babbit is then scored at the joint to facilitate removal of the cap for the removal of the paper shims and of burrs from the joint faces of the rod and cap.

After the re-assembly, the oil grooves are milled in the big end bore, following which both sides of the big end boss are machined and a chamfer is formed in the big end bore. Control of weight to very close limits is important, and at this stage two operations are performed to ensure that the specified weight tolerances are maintained. At the first, the small end boss is hollow milled to a tolerance of 2 grs and at the second, the same tolerance is maintained on the large end by hollow milling both sides of the boss to whatever depth is necessary to give the specified weight.

A Diskus Werke surface grinder, see Fig. 6, is used for the next operation, grinding the faces of the big end boss. The machine table carries two different work-holding fixtures arranged alternately. A rod is mounted in one fixture with location taken from the centres and is traversed first under a roughing and then under a finishing grinding wheel. At the loading station, the rod is then transferred to the next fixture with location taken from the ground face so that on the second traverse the other side is first rough and then finish ground to width. The work-holding clamps are applied and released automatically. Each work-holding device incorporates a roller follower mounted in an arm that is free to pivot. Opposed to the roller follower is a spring-loaded plunger. At the loading station, the follower contacts a cam track so formed that the spring-loaded plunger is forced in to allow the clamps to come clear of the work. When the follower leaves the cam track, the plunger is forced out and the clamps are applied.

There are two further machining operations to complete the machining sequence. At the first of these, the big end and small end bores are fine bored to size on an Alfing boring machine, see Fig. 7. This machine is arranged to take six rods and to bore three alternate rods simultaneously. The practice is, of course, to load three rods while three are being machined. Location is once again taken from the centres. The centres on one side of the

Fig. 7. Alfing machine for fine boring large and small bores

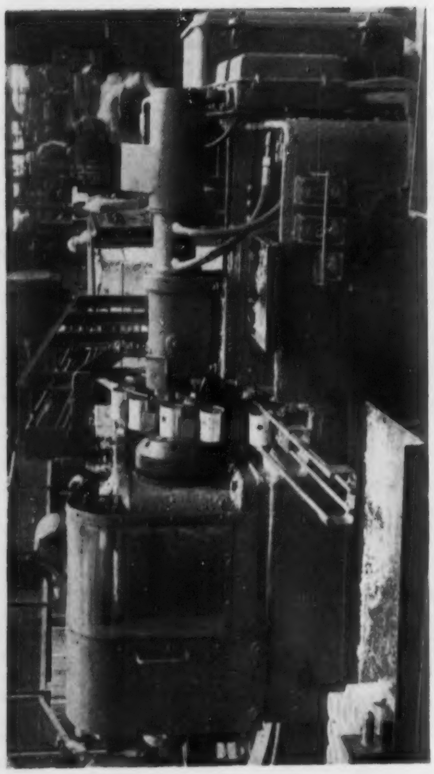


Fig. 8. Special Ford machine for machining location register on pistons

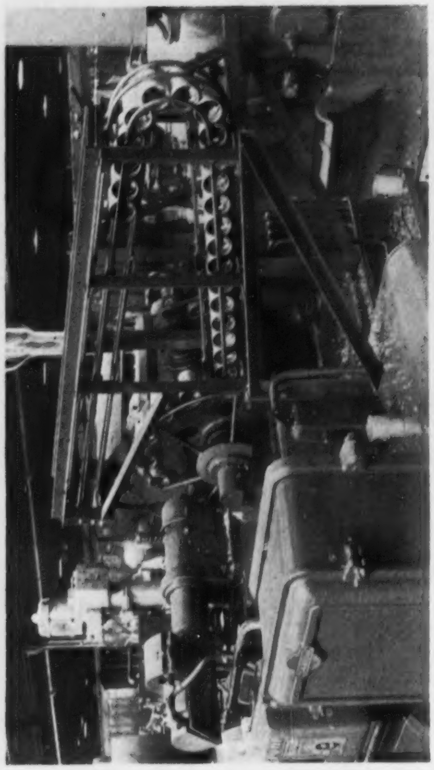


Fig. 9. The delivery chute for the machine shown in Fig. 8

Fig. 10. Set-up for machining O.D., grooves and crown on a six-spindle Baird automatic

Fig. 11. The work-holding fixture on the fine boring machine for gudgeon pin holes

rod are fixed, those on the other are spring-loaded. The machine is completely automatic. While the machine cycle is being carried out on three rods, hydraulically actuated wedges retract the spring-loaded centres for the other three stations. At the completion of the first cycle, the wedges are automatically retracted from the stations holding the rods to be machined and the spring-loaded centres advance to hold the work securely. At the same time the moving centres are retracted from the rods that have been bored and the table moves transversely to bring three more rods into the boring position. A reaming operation on the small end completes the machining.

Before transfer to the engine assembly line, a very exhaustive inspection sequence is carried out. To begin with, every small end bore and every large end bore is checked for diameter by means of special Mercer gauges; this is followed by 100 per cent check of the thickness of babbit in the large end bore by a special electrical testing device arranged to indicate the thickness of babbit directly on a dial. Readings are taken at several points round the bore to ensure that the maximum and minimum thicknesses conform to specification. Finally, every rod is checked for alignment, squareness of side faces and parallelism of large and small bores on an acceptance fixture.

Fig. 12. Special Ford machine for machining circlip grooves

Piston production

For the Ford Anglia and Prefect, autothermic full skirt pistons in low expansion aluminium alloy are used. In accordance with universal practice in piston production, the first machining operation is carried out to give locations for subsequent operations, that is, the skirt end is faced in relation to the inside face of the crown and a location register is bored in the skirt. Although this is conventional practice, the actual methods employed are far from conventional.

A special boring and facing machine of Ford design and manufacture is used. It is illustrated in Figs. 8 and 9. Before the introduction of this machine, the operator had to load a piston into a chuck, tighten the chuck, start the machine cycle, release the chuck and unload. Even with an air-operated chuck, the idle time was high in relation to the actual machining time.

With the new machine, the operator feeds castings into a chute, which is so designed that it is impossible to insert the casting incorrectly. The chute, which is in approximately the form of a "U" placed horizontally, is loaded from the front of the machine. The piston rolls by gravity down the first leg of the chute to the back of the machine and then by gravity down the second leg towards the front of the machine. At the end of the chute, the piston is picked up by a toothed carrier wheel, which is arranged to revolve in synchronization with the machine cycle. Eventually the casting is carried round until it is in line with the machine chuck. At this point a ram comes forward and pushes the casting into the chuck, the boring bar advances to face the skirt and bore the register. At the end of the machine cycle, the piston is automatically pushed clear of the chuck, the carrier wheel makes a part revolution and the piston rolls down a delivery chute to the operator's position.

The second operation is carried out on a Baird six-spindle automatic, see Fig. 10. Location is taken from the machined face of the skirt and from the bored register. Loading is effected at the first station. At the second, the O.D. is rough turned part way, the crown is rough faced, and the three ring grooves are formed to depth. Incidentally, the casting is so designed that the oil return slots in the bottom ring groove break through at this stage, thus obviating the necessity for a subsequent operation solely for machining these slots.

At the third station on the Baird machine, the remainder of the O.D. is rough turned and the land diameters are semi-finish turned. The fourth station is tooled to finish face the crown and to chamfer the ring grooves. At the fifth station, the skirt and the land diameters are finish turned and at the sixth and final station the ring grooves are formed to width.

A No. 2 Cincinnati centreless grinder is used at the next operation. It is

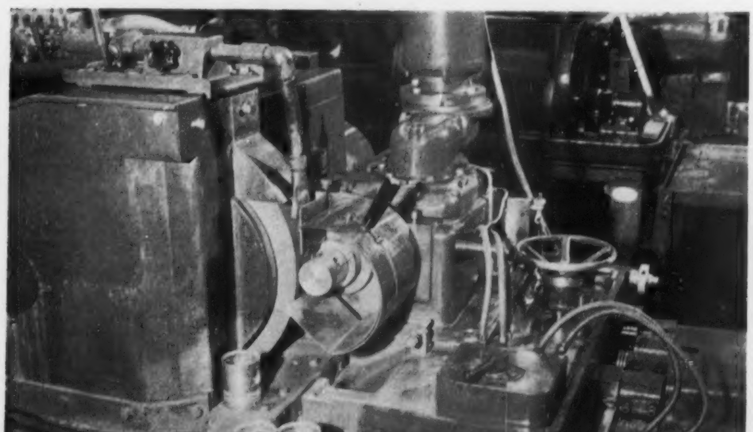


Fig. 13. The Scrivener cam grinder for producing the piston skirt contour

arranged for plunge grinding to rough grind the skirt and the land diameters to a 0·002 in tolerance. The piston then passes to another Cincinnati centreless grinder, also arranged for plunge grinding On this machine, the skirt is semi-finish ground and the land diameters are finish ground.

The gudgeon pin holes are fine bored on a double-end Ludwig Burger fine borer, see Fig. 11. This machine is arranged to bore three pistons simultaneously. Each of the pistons is dropped into a pot-type fixture with the gudgeon pin holes approximately in line with the machine spindles. Actuation of the clamping device first causes locating fingers to move and bring the pistons into correct alignment with the machine spindles. Should any piston have been loaded so far out of alignment as to make it impossible for the locating finger to correct the error, the machine remains inoperative. As soon as alignment has been effected, the clamping head comes down and clamping pads move out to contact the bore of the skirt. One end of the machine is tooled for rough boring and the other for fine boring the pin holes to very high standards of accuracy and finish.

An unusual machine of Ford design and manufacture is used to machine the circlip grooves. The loading position for this machine is shown in Fig. 12. The piston is mounted on a sub-table with location taken from one gudgeon pin hole. There are four stations with a form tool mounted above the piston at each station. As the main table revolves and the work is carried away from the loading station, the sub-table carrying the work automatically starts to revolve and the form tool is fed down to it.

Before the next major operation, an extremely important one at which the piston skirt form is generated on a Scrivener cam grinding machine, the register is remachined and the skirt is faced in relation to the O.D. and the gudgeon pin holes. This Scrivener machine is shown in Fig. 13. Essentially, the general principle employed is that the work is rotated about a horizontal axis and is at the same time given a controlled oscillation in a horizontal plane. A normal Scrivener grinding head is used, but there is a special workhead.

The driving spindle on which the piston is carried is mounted in a head that is arranged to pivot about a fixed point. Mounted on the spindle at the end remote from the work there is a cam of appropriate shape. This cam is maintained in contact with a fixed segmental follower by means of a spring. The cam has two lobes, each lobe coinciding with the axis of a gudgeon pin hole. The diameter ground immediately above the pivoting point is a true circle; as the distance from the pivot increases so does the effect of the cam action increase.

The pivoting workhead housing is carried on a main table driven by Scrivener controlled-cycle mechanism. It is the forward position of the table that determines the final dimensions of the work. The controlled cycle mechanism is actuated by a cam arranged to give rapid advance, in-feed and rapid retraction in a completely automatic cycle. The complex skirt form must be produced to very close tolerances and frequent checks, involving measurements at 5 deg intervals, are carried out.

To complete the machining on the piston, the gudgeon pin holes are reamed on a special Ford machine with a special expanding reamer. The piston is mounted crown down in a pot type holder, mounted on the machine table, and is fully floating. A reamer of special form is used. It has a slight camber along the length of each cutting edge. Reciprocating motion is imparted to the work through a hydraulic system. This operation removes about 0·0002 in on the diameter and develops a surface finish in the order of 2 to 3 micro-inches r.m.s.

CORRESPONDENCE

FRONT SUSPENSION

SIR,—With reference to an article on divided axle front suspension systems, by D. R. Hume in your January issue, I should like to correct certain errors contained therein, and point out one disadvantage of these systems that the author has omitted.

First, the author states that the wheel tilt, or camber change, with deflection is greater in the case of a low pivot centre than in that of a high centre. However, the camber change is determined solely by the deflection and the horizontal distance 'd' from the pivot to the centre of the tread. In fact, it is to a close approximation, $\theta = \tan^{-1}\left(\frac{x}{d}\right)$, where 'x' is the vertical deflection of the wheel.

Secondly, in the drawing of the geometry of the Ford suspension, Fig. 6, the upper of the pair of lines locating the swing centre of each wheel should be perpendicular to the king pin, and not horizontal as shown. This will result in a slightly higher roll centre and shorter swing radius than is shown in the illustration.

These are, however, only minor points, whereas a major fault in an otherwise informative article is, I feel, the omission of any consideration of the "sprag effect," which can be of considerable importance in suspension systems of this type. This effect, causing an upward force to act through the independent suspension system and lifting that end of the car when subject to a lateral acceleration, occurs with any independent suspension system with a roll centre above the ground level. In the case of a swing axle system, this upward movement gives rise to a large, positive wheel camber. Since, because of the weight transfer, the outside wheel provides most of the cornering force, this camber change results in a loss of cornering power, and in most practical cases gives a marked "roll understeer" characteristic which is, to my mind, both excessive and undesirable. With a swing axle rear suspension, of course, an oversteer characteristic usually results.

The magnitude of the sprag effect may be shown from the Allard Palm Beach data supplied by Mr. Hume. Assuming a lateral acceleration of 1 g, the upward force on the front end of the car is approximately 300 lb. This causes the front of the car to rise 2·5 in, thereby increasing the positive camber on the front wheels by approximately 6 deg. As a result, the outside wheel leans over at 8 deg from the vertical, compared to 5 deg for the car with equal length horizontal links, and only 3·5 deg for the car with equal and parallel links and a roll centre of 11 in. The resulting marked understeering characteristic may be of some value when a large proportion of the weight is carried on the back wheels as in the J2 Allard, but where, as in most modern cars, the preponderance of the weight, unfortunately, is on the front wheels, it is hard to see any real virtues in the single link system other than that of mechanical simplicity. A double transverse wishbone layout can be designed to incorporate all the virtues of the divided axle with none of its vices, and in particular can be arranged to eliminate the sprag effect no matter how high a roll centre is desired.

JOHN E. JACKSON, M.Sc.

New Film

A NEW film entitled "They've Come a Long Way" has recently been shown for the first time in London. In this film, which was produced by the Orion Picture Corporation, the development of Garringtons Ltd. is depicted. It begins by telling how John Garrington started in a small way as a blacksmith near Darlaston, in Staffordshire, in 1837. In 1840, he opened his first factory, called Prince Albert Works, which consisted of one office and one small shop. Then the film goes on to describe the growth of the company to its present-day position, and shows scenes from their Newton Works, which cover about 75 acres at Bromsgrove, Worcs. This film will shortly be available in 16 mm and 32 mm sizes.

ENGINE OILS

S.A.E. Numbers and Their Significance

By R. H. Warring

CLASSIFYING motor oils according to "grade" is an arbitrary practice since the specification adopted for any nominal grade may vary considerably from one manufacturer (oil refiner) to another. The designations light oils, medium oils, heavy oils, etc., are therefore largely meaningless as a satisfactory basis for comparing grades produced by different refiners. They are, in fact, of no more use than the name, number or letter code originally used by individual manufacturers to classify the range of oils of any particular brand.

Probably the most satisfactory world standard of classifying motor oils is according to the S.A.E. number system where the numbers used do relate to the viscosity of that particular oil, viscosity being an important criterion as regards the performance of a motor oil. It must be appreciated, however, that classification by S.A.E. number is concerned only with viscosity and takes no account of quality or other factors (except in the case of S.A.E. numbers of 75 and above, appropriate to gear oils, which also take into account consistency or "channelling").

Thus, S.A.E. classification is still, largely, an arbitrary method since, with the exception of extreme pressure gear oils, no account is taken of other major factors which may affect performance, such as the use of inhibitors and other additives, etc. Its main use, therefore, is in providing a set of fairly loose standards from which approximate comparisons may be drawn. The ultimate performance of any oil can only be assessed by lengthy laboratory tests and running tests under service conditions.

The present S.A.E. numbers represent the third major attempt by the American Society of Automotive Engineers to introduce a satisfactory, unified standard for motor oils. Introduced in 1911, the first series of S.A.E. classifications took into account viscosity, flash point, pour point, carbon residue and a number of other factors. A second series, introduced in the mid-1920's, was based on viscosity in ten "name" stages. This system became meaningless when various refiners used the same or similar designations for their branded products with no reference, or even in contrast, to the S.A.E. viscosity scale. The present system, which was first started in 1926, classifies oils by viscosity again, but this time using S.A.E. numbers. This is still an arbitrary classification but, being based on viscosity, has become an acceptable world-wide standard. The range of S.A.E. numbers has been increased from time to time, with the basic limits revised and brought up to date, as necessary.

Basically the viscosity of an oil—or any fluid, for that matter—is a measure of its internal friction or resistance to flow. The "thicker" the oil, in popular language, the higher its viscosity and thus its greater reluctance to flow (that is, it has greater internal friction). Viscosity is probably the most important single factor associated with the characteristics of lubricating oils.

The absolute viscosity of a fluid is a function of the shear stress in the liquid, measured parallel to the direction of flow, and the rate of change of velocity of flow as the distance from the boundary wall increases. In working terms, absolute velocity can be defined as the force necessary to move a plane surface of unit area over another plane surface at a specified speed when the two surfaces are separated by a layer of fluid of unit thickness. When the units become one square centimetre, one centimetre per second and one centimetre, respectively, the resulting force required is termed a poise.

Poise is a measure of absolute viscosity and is largely of laboratory interest only. For engineering purposes it is the kinematic viscosity associated with dynamic flow of the fluid that is of significance. Kinematic viscosity has a simple relation to absolute viscosity, being equal to absolute viscosity divided by the specific gravity of the fluid at the temperature at which the viscosity is measured. The unit for kinematic viscosity is the stoke. Since this leads to fractional viscosity values for a large number of liquids, a smaller unit, the centistoke ($\frac{1}{100}$ of a stoke) is commonly used in practice. Similarly, absolute viscosities are often quoted in centipoises.

For the practical determination of viscosities various standard instruments are used. In this country the Redwood 1 or Redwood 2 instrument is probably the most common (the latter being designed to handle more viscous liquids). The Saybolt scale is most common in the United States and the Engler viscometer in Continental Europe. All these instruments work on the same principle, in that they determine the time in seconds in which a known volume of fluid will flow through an orifice under the action of gravity. The kinematic viscosity can then be calculated from the formula:—

kinematic viscosity $= Ct - D/t$

where C = calibration constant for the instrument.

D = design constant for the instrument.

t = time of flow, in seconds.

The constants are normally arranged so that the resulting kinematic viscosity is calculated in centistokes. Distilled water, having a known kinematic viscosity of 1·0038 c.s at 68 deg F, is used for determining the calibration standards. Table I gives the conversion of kinematic viscosities to practical scales.

The viscosity of any fluid varies appreciably with temperature. As temperature increases, viscosity decreases,

TABLE I. CONVERSION KINEMATIC VISCOSITIES TO PRACTICAL SCALES

Kinematic viscosity centistokes	Redwood No. 1 seconds at 70 deg F	Saybolt Universal seconds at 100 deg F	Engler deg
10	52	58·8	1·83
15	68	77·2	2·32
20	85	97·5	2·9
25	104	118·9	3·45
30	123	140·9	4·1
35	143	163·2	4·7
40	164	185·7	5·35
45	184	208·4	6·0
50	204	231·4	6·65
55	224	254·4	7·25
60	244	277·4	7·9
65	264	300·4	
70	285	323·4	over 60 :— multiply by 0·132
75	305		
80	325	over 70 :— multiply by 4·620	
85	345		
90	366		
95	386		
100	406		
	over 100 :— multiply by 4·06		

TABLE II. S.A.E. NUMBERS SPECIFICATION

	S.A.E. Number	Viscosity Saybolt At 0 deg F min.	max.	Universal seconds At 210 deg F min.	max.	Lower temperature limit for no channelling
Crankcase oils	5W	—	4,000	—	—	—
	10W	6,000	Not above 12,000	40	—	—
	20W	12,000	48,000	45	—	—
	20	—	—	45	less than 58	—
	30	—	—	58	less than 70	—
	40	—	—	70	less than 85	—
	50	—	—	85	110	—
Gear Oils	75	—	15,000	—	—	−40 deg F
	80	15,000	100,000	48	—	−20 deg F
	90	—	750,000	75	120	−10 deg F
	140	—	—	120	200	+20 deg F
	250	—	—	200	—	—

hence the viscosity at operating temperatures is of particular significance in the case of motor oils. The ideal is an operating temperature viscosity high enough to maintain a satisfactory fluid film for lubrication, but low enough to avoid excessive fluid friction. Friction would, in any case, raise the temperature of the oil still further, and also the temperature of the bearing surfaces.

S.A.E. viscosity numbers are determined with reference to the same Saybolt Universal scale at one or two temperatures — 0 deg F and 210 deg F. Limits are fairly wide, specifying a maximum and minimum viscosity at 0 deg F in the case of the thinnest oils and a maximum viscosity at 210 deg F for the thicker oils, with an overall proviso that the viscosity of oils for use in crankcases should not be less than 39 (Saybolt Universal) seconds at 210 deg F. This corresponds, roughly, to 3·6 centistokes or about 34 seconds Redwood standard (1·28 deg Engler).

Gear oils are classified in a similar manner, with maximum and minimum limits for viscosity (Saybolt Universal seconds) together with consistency requirements at low temperatures. Channelling is a property of thick oils and greases which when particularly thick (for example, such as in cold weather) causes a part line or "channel" in the body of the lubricant so that the surfaces or parts to be lubricated may actually work without coming into full, or even partial, contact with the lubricant.

Specifications for the full range of S.A.E. crankcase and gear oil classifications are given in Table II, and the appropriate positions and relevant particulars for typical British branded motor oils in Table III. Although, as pointed out previously, classification by S.A.E. number takes no account of the quality of the oil, it can be assumed that the table of branded oils gives "equivalents" in the various brands which may be freely interchanged.

It must be clearly appreciated that some first grade motor oils contain additives which increase the detergency of the oil; these oils have the effect of loosening accumulated deposits of sludge in engines which have been run on other types of oils. It is particularly important, therefore, to take any precautions necessary to remove deposits loosened in this way when changing over from a non-detergent oil to a detergent grade.

A further point of interest is the viscosity index of a motor oil, although this is seldom specified except in more technical literature. Broadly, the viscosity index is a measure of the oil's viscosity-temperature characteristics. It is of particular significance where oils are required for applications that involve operation over a wide range of temperatures. The higher the viscosity index the smaller the change in viscosity with changing temperature. The viscosity index of any oil can be calculated if its viscosity at two temperatures is known. For viscosity index "standards" a typical naphthenic oil has a viscosity index of 0 and a typical paraffin a viscosity index of 100. Owing to developments in the refining process, the use of additives, etc., many modern oils have a viscosity index in excess of 100. In other words, as far as engine oils are concerned, modern oils approach more nearly the ideal of a lubricant which has a relatively low viscosity for easy starting from cold whilst maintaining the same viscosity at elevated working temperatures for adequate lubrication and minimum oil consumption.

TABLE III. SELECTED LIST OF REPRESENTATIVE COMMERCIAL OILS

GRADE

Shell X-100	10	20	30	40	50
S.A.E. No.	10	20/20W	30	40	50
Specific gravity	·870	·880	·883	·886	·890
Viscosity index	95–100	95–100	95–100	95–100	95–100

Esso	10	20	30	40	50	Racer
S.A.E. No.	10	20	30	40	50	50
Specific gravity	·875	·881	·885	·888	·890	·891
Viscosity index	98–100	98–100	98–100	98–100	98–100	102

Energol	10	20	30	40	50
S.A.E. No.	10	20W	30	40	50
Specific gravity	·879	·885	·890	·895	·900
Viscosity index	95–100	95–100	95–100	95–100	95–100

Mobiloil	Arctic	A	BB	D
S.A.E. No.	20	30	50	50
Specific gravity	·880	·895	·895	·895
Viscosity index	100 (min.)	92	92	95

Castrol	Castrolite	XL	XXL	Grand Prix
S.A.E No.	20	30	40	50
Specific gravity	·885	·890	·895	·897
Viscosity index	114	92	92	92

Havoline	10W	20W	30W	40W	50W
S.A.E. No.	10	20	30	40	50
Specific gravity	·870	·879	·886	·889	·894
Viscosity index	105	99	94	95	91

DRIVING SEATS

An Appraisal of the Mechanical Basis of Comfort

By G. E. Cleaver

UNTIL the scientists can offer a more adequate definition of comfort, the design of seats is likely to remain as much of an art as an engineering exercise. Many attempts at a scientific approach have been made but subjective judgment has never substantiated the claims made for the results. The problem is aggravated by misconceptions of buyers on the characteristics of a comfortable seat. Many think that softness is synonymous with comfort and expect to judge a seat by merely sitting on it. While the laws of seating may be ignored temporarily by this method, they eventually manifest themselves most painfully on a long journey and judgments have to be revised.

A considerable amount of work has been done by anatomists in the field of postural mechanics and the resulting data are invaluable to the seat designer. Unfortunately, as discussed later in this article, the results are not sufficient to establish general principles and the seat designer must still interpret them for himself and relate them to whatever measurements of the characteristics he can obtain from seats of accepted comfort. Therefore, while any claims of 'scientific' seating must still be viewed with doubt and suspicion, a more engineering approach can be quite practical and most advantageous.

The present article attempts, firstly, to describe the mechanical functions of a seat and the loads to be carried. The relevant anatomy and postural mechanics are then explained and the distribution of the loads in each of the various postures described. Some of the faults of design are mentioned, and in conclusion the prospects of developing standard data for the future are considered.

The mechanical function of the driving seat

While an adequate definition of comfort is difficult to find, it seems valid to describe its mechanical basis as one in which the desired posture is maintained with the least amount of muscular effort and with the pressures involved distributed in a manner in which the human form is best able to carry them. To complete such a description, a dimensional specification of the desired posture is needed, a means of measuring muscular effort needs to be available and a physiological description of how pressures are best carried would have to be provided. For the practical purposes of seat design, however, there are sufficient anthropometrical data and the means of measuring the load/deflection characteristics of existing seats available for subjective tests to be confirmed and analysed until more scientific means allow the description to be more adequately defined.

In the sense given above, the mechanical function of a driving seat may be defined as 'to support the occupant in the driving posture in a condition of stable equilibrium by exerting forces equal and opposite to the loads imposed by his weight.'

The driving posture is merely one of many which are adopted by humans in following the course of modern life, and others—of writing, typing, dining, conversing and resting—come to mind, each of which determines the shape of the seat to be provided. The appropriate state of mind must also be considered. We can be listening to a conversation in an easy chair and, at the same time, be enjoying the comfort afforded by a well-sloping back. But, when a point of argument is reached, we invariably sit up and adopt an entirely different posture—less comfortable, but more appropriate to our state of mind. In driving, an attitude of alertness is needed and it is quite likely that the seat with a well-sloped back which provides the most comfort will be uncomfortable for the driver who needs to take a more upright posture to conform to his mental attitude of alertness. It is, perhaps, in this conflict between 'mechanical' comfort and 'mental' comfort that many scientifically designed seats fail to find subjective confirmation for their merits.

The 'shape' of the driving posture is determined largely by the position of the controls and it is obvious that a large degree of the comfort attainable in a seat is determined by the relative position of the controls. The controls, however, determine only the posture of the arms and the lower legs, and the forward vision determines the posture of the head. If, therefore, the control and screen positions result in well-placed shoulders and knees, it is for the seat to carry the trunk and the upper leg correctly for maximum comfort to be attained. In other words, although the components of the posture are closely related, they can be clearly divided into seat position and seat shape.

This division is particularly important in considering the range of sizes of drivers. The differences must be accommodated almost entirely by the positioning of the seat and not by the shape of the seat because the anatomical dimensions concerned with the shape vary very little and are easily accommodated, as will be shown later. It would be faulty design to attempt to correct errors in control positioning by adjusting the shape of the seat to suit.

For present purposes, the mechanical function of a driving seat, as described above, may be limited to 'the support of the trunk and the upper legs of the occupant in the driving posture in a condition of stable equilibrium by exerting forces equal and opposite to the loads imposed by the weights involved' (that is, all the weight of the body less the amount carried by the floorboard at the heels). If this can be done without involving any muscular effort then the seat may be considered to provide 100 per cent comfort—at least as far as it is determined by mechanical considerations.

The mechanical functions of upholstery

Most of the research on anatomical seat design has been directed toward hard, unupholstered seats. When upholstered seats have been dealt with, only the shapes in the fully compressed condition have generally been specified and little consideration seems to have been given to the behaviour of the upholstery in moving from the free shape to the occupied shape. It is undoubtedly a most complex mechanical system but the comfort of the seat depends heavily upon the load/deflection characteristics and the opportunity is taken here to attempt to make good the neglect and to draw more attention to their importance.

The load/deflection characteristics of an upholstered seat are covered by the three phases, reception, ride and resistance. Reception, the first phase, provides comparatively large deflection under a range of postural loads which are distributed over the areas of support, and

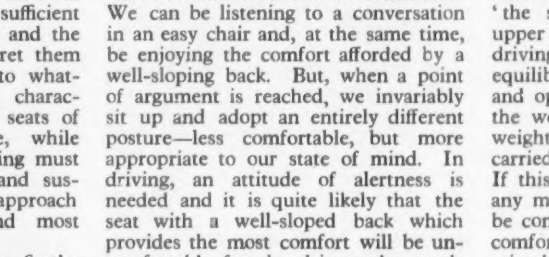


Fig. 1. Load/deflection characteristics of upholstery

extends from the free shape to the postural shape. The second phase is 'ride' in which the upholstery absorbs the shocks passed through the seat when the car is traversing bumpy roads. This phase extends from the statically loaded position to the position under the load imposed by the normal maximum acceleration used in the design of the suspension system of the vehicle. It is essential for maximum comfort in this phase for the postural trunk shape to be maintained and the posture of the upper leg altered only by a slight pivoting movement at the knee joint. To avoid bouncing it is also desirable for the upholstery to possess as much damping as possible. The third phase, resistance, is one of 'steep' deflection in which abnormal bumps are absorbed in a short distance without bumping through. These three phases and their load/deflection characteristics may be shown diagrammatically as in Fig. 1.

During reception, the cushion wraps round the legs slightly and offers lateral support. The loads on the squab are too low to allow this except with very soft upholstery and the extra support offered by the shaped bucket seat has much in its favour. Excessive deflection is most undesirable because the extension of the contacting areas beyond that which can carry vertical loads only causes unnecessary movement between the seat cover and the occupant's clothes. It also causes serious reduction in the circulation of air by which excessive body heat is dissipated.

The extent of the reception deflection is determined, of course, by the difference of the free shape of the seat from the occupied shape. To avoid stresses which tend to stretch the seat covering and to burst its seams, it is desirable to keep the length of the covering the same in the free shape as it is in the occupied shape. Much of the desirable load/deflection characteristics of upholstery is often lost by restrictions caused by the covering.

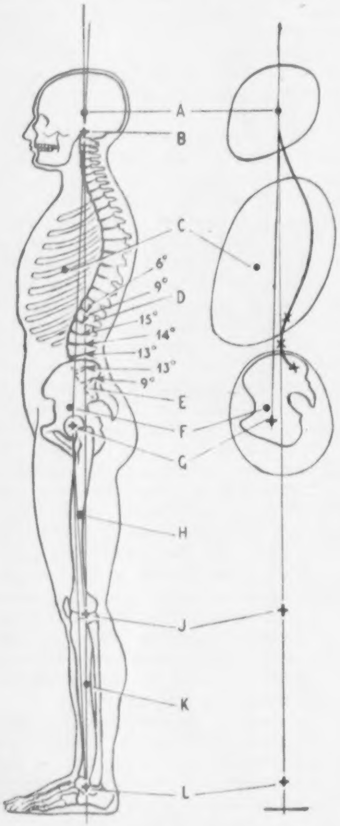
It is evident that a free shape differing largely from the occupied shape will require softer initial deflection than one that conforms closely. For the sake of a tidy appearance a tensioning of the covering is unavoidable and this can only be obtained over a convex shape which is deflected under the load to the concave shapes accommodating the rounded human form. The determination of the free shape will always, it seems, be a compromise between the load/deflection characteristics required and all the other factors concerned with attractive appearance, low stressed covers, etc.

It will be seen that the idea that softness is synonymous with comfort is a delusion. A soft seat certainly *feels* comfortable because of the even distribution and low unit pressure which it affords, but, after a short while, its shortcomings are painfully revealed by aching muscles. Initial softness is certainly most desirable as far as a tidily stretched cover will allow. But after deflection has ceased and the weight of the body is counterbalanced by the increased build-up in the upholstery, it is *resistance* that is required to support the body in the desired posture, not the continued yielding to an indefinite degree of envelopment by soft upholstery.

Through the range of movement, therefore, we are concerned with shapes and pressures and the two are closely related. It is not only necessary for the seat to provide forces equal to the loads but for the forces to act suitably when deflection to the postural shape has been reached.

The anatomy of the human body

While the weight of the human body is carried through 'Nature's own upholstery'—the flesh, it is transmitted down the bones and the skeletal system needs to be studied closely if the distribution of the loads is to be found and the equal and opposite supporting forces are to be designed into the seat.



A —Centre of gravity, head.
B —Atlas
C —Centre of gravity, upper trunk
D —Lumbar/upper trunk joint
E —Lumbar/pelvis joint
F —Centre of gravity, lower trunk
G —Hip joint
H —Centre of gravity, upper leg
J —Knee joint
K —Centre of gravity, lower leg
L —Ankle joint

Fig. 2. Anatomical diagram (left) and diagram of forces (right) for the standing posture

In the anatomists' definition, the skeletal system includes all the cartilage holding the bones together and the ligaments which pass over the joints as well as the bones themselves. There are limits to the extension and flexion of all joints, and it is obvious that the components must lie well between these limits if a posture is to be a comfortable one. The ability to hold an erect posture is only a comparatively recent attainment of man and the smart straight back adopted by the Guards and by the well-trained horseman lies very close to the limit of backward bending at the hips. In the correct driving posture, all joints lie approximately midway between the limits of their movements and it is only in dealing with support for the lower spinal joints that close attention to flexural limits is involved.

For this discussion, the human body may be conveniently divided into the trunk, the head, the upper legs, the lower legs and the arms. As already pointed out, all must be considered in determining the seat position and the postural weights. But only the trunk and the upper legs need be considered when the seat shape is to be determined. The skeletal framework of the trunk is composed of the spine, the pelvis and the ribs, as shown in the standing posture in Fig. 2 (left). For the present purpose, the spine and pelvis can be considered as one system, and the ribs ignored.

The spine is composed of thirty-four vertebræ, twenty-four of which are flexible in varying degrees; seven in the neck, twelve in the upper back and five in the lumbar. Five of the immovably-jointed bones form the sacrum and the remaining five the vestigial remains of our ancestral tail. The sacrum is immovably jointed to the hips and haunch bones to form the pelvis. The spine and pelvis are shown overlined in Fig. 2.

In which direction support must be offered to enable the system to act as a column in transmitting the weight of the upper part of the body to the cushion is determined by the flexible nature of the spinal joints and the rotary movement of the pelvis around the hip joint. The degree of movement in the various regions of the spine may be judged by some maximum amounts recorded between backward and forward bending that are indicated in Fig. 2. The amount of movement between the vertebræ in the lower back above those mentioned is negligible. While the edges of the bones and the ligamental sheath limit the flexing, it is, of course, weight and the action of muscles that maintain the spine in any one position. The region where the relative movement between the vertebræ is greatest is where support can best be applied to relieve the muscles.

Postural mechanics

The Standing Posture. In the standing posture illustrated in Fig. 2, the centres of gravity and the joints of each of the main components are shown. The earlier work of anatomists led them to assume that all the centres of gravity

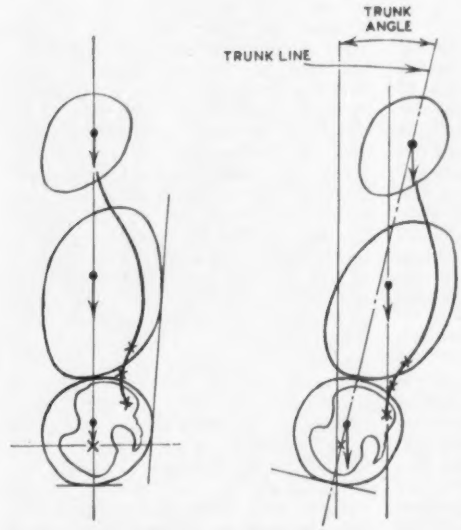


Fig. 3. Diagram of forces for the erect seated posture (left) and the fully-rested seated posture (right)

and joints lie on the same vertical line. Later research, however, has shown that such a posture would require a considerable amount of muscular 'trimming' because of the lack of stability. The posture which can be held for the longest duration with the minimum amount of fatigue is where the hip joint lies slightly ahead of the gravity line and the restricting ligament allows the joint to act as a toggle. Similarly the knee joint is 'toggled' behind a straight line running from the hip to the ankle.

However, the anatomists have overlooked the effect of the articulation of the spine. While the mid-lumbar joints lie very close to the gravity line passing through from the neck to the ankles, the other spinal joints do not, particularly the lumbar/pelvis joint. There are moments at this joint which tend to rotate the pelvis around the hip joint and which need to be counteracted by muscular or ligamental stress.

That all joints lie on the same vertical line in the standing posture to give a condition of equilibrium is not, therefore, a valid conclusion. The use of one centre of gravity for the trunk is also invalid. In fact, the assumption that the postural mechanics for the standing posture apply to the sitting posture has often vitiated many attempts to design 'scientific' seats; the recognition of the mechanics of the spine and pelvis is essential to seat design if true progress is to be made.

Fig. 2 (right) shows a diagram of forces that simplifies the joints in the lumbar region by assuming a toggle joint in the mid-lumbar region, a pivot point for the upper trunk at the upper lumbar joint and a pivot point at the lumbar/pelvis joint. A centre of gravity is shown for the lower trunk and another centre of gravity for the upper trunk (the effect of the weight between these two masses is ignored). It will be seen in this arrangement that there are moments of the upper trunk around the upper trunk/lumbar joint which require the fullest consideration if the prototype seat is to fulfil the designer's intentions.

Before dealing with the mechanics of driving posture it is convenient to deal first with an erect seated posture and then with a sloping posture in which the back can be fully rested.

The Erect Seated Posture. The erect seated posture is defined as one in which a vertical line drawn tangentially through the centreline of the neck vertebræ passes through the point of support on the haunch bones as shown in the diagram of forces, Fig. 3 (left). In the action of sitting, the pelvis rotates with the hips. As the backside reaches the seat, the weight of the body is taken on the haunch bones and the point of contact becomes the pivoting point of the pelvis. The rotation throws the lumbar/pelvis joint further back so that the moments around the rotating point are larger than in the standing posture. The amount of muscular and ligamental effort required to hold the spine firm is, therefore, greater in the erect seated posture than in the standing posture. It will be seen that all the component weights tend to push the body either downward or forward. While there are no forces backward, no backrest can be of any use in this posture because there are no forces against which it can offer 'equal and opposite forces' to provide stable equilibrium. The fallacy of providing backrests for an erect posture is common in many forms of 'scientific seating.'

Where a forward support is provided, however, such as the steering wheel in the driving posture, much of the forward weight can be taken through the arms. Any back support which may be gained in this way for holding the lumbar region firm and the pelvis against rolling must, of course, be balanced by extra stress in the arms. As no back support can be offered for an erect seated posture, the support of the spinal joints, resistance to the rotation of the pelvis and the upright carriage of the upper trunk must all be obtained by either muscular or ligamental action.

The Fully-Rested Seated Posture. The mechanical requirements for supporting all the loads imposed by the seated posture becomes apparent from an analysis of the loads involved in the erect seated posture. The back must be inclined sufficiently so that all the component loads of the trunk and pelvis are either downward or backward of the point of support so that they may all be supported by either the cushion or the squab. The centre of gravity of the upper trunk must be vertically above or behind the lumbar/pelvis joint to avoid forward moments, and the lumbar joints supported so that the upper trunk load can be transmitted through to the point of support without flexure. This condition is illustrated in the diagram of forces for the fully-rested posture in Fig. 3 (right).

The erect posture previously described is one in which the back is braced by muscular action and in which the bottom points of the haunch bones happen to coincide with the point of support. If muscular stress is released, the pelvis will rotate around the hip joint without altering the trunk line because the point of support remains virtually the same.

The vertical line of the erect sitting posture can, therefore, be adopted as a trunk line and used for all types of sitting posture. The degree of reclining of any posture can be measured by the backward inclination of the trunk line and the angle termed the trunk angle.

While the trunk load, as far as it is carried by the cushion, is concentrated at the haunch bones, the introduction of a trunk angle introduces a resultant which tends to slide the backside forward along the cushion and must be fully resisted in the design of the cushion if equilibrium is to be obtained. The vertical trunk load, with a supported lumbar, is transmitted at the trunk angle and a resultant force acting backward is required. The lumbar supporting forces also create similar resultants.

The cushion must, therefore, provide pressure under the thighs when depressed and the angle and forces of the depressed cushion in relation to the angle and forces of the depressed squab are obviously important.

Friction between the cushion cover and the occupant's clothing can offer

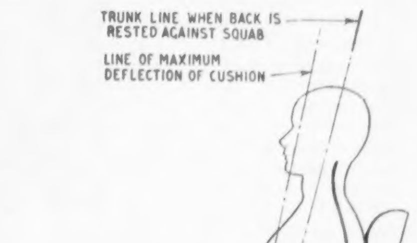


Fig. 4. Diagram showing lack of coincidence between most comfortable position against squab and most comfortable position on the cushion

some resistance to the resultant but this is inevitably accompanied by 'strangulation' of the crutch; easement from which is a common and frequent necessity on many a long journey in many types of cars.

Little so far has been said about the support of the upper leg. While the weight of the foot and lower leg is taken by the floorboard and toeboard, the angle at which the upper leg will lie is determined by the position of the point at which the heel rests relative to the position of the haunch bones on the cushion. If the cushion supports more than the weight of the flesh underneath the thigh bones then its forces will tend to act against the forces offered by the floorboard. It may be thought that it would relieve some of the weight off the heel but it is obviously not an efficient point of leverage and there is a risk that the pressure might restrict the arteries and press upon the nerve cords running through the underside of the thigh. It may also interfere with pedal operations.

It is apparent, for maximum comfort and minimum supporting loads, that the resistance must be offered within a comparatively short distance forward of the point of haunch bone support.

The Driving Posture. In considering the driving posture, in light of what has been shown for the fully rested posture, the dynamic forces must also be included. The alertness demanded for driving under varying conditions, town and country, clear vision and fog, fast and slow speeds, will influence the driver's posture and an optimum angle can only be decided by experiments.

Braking and acceleration will introduce forces which are best resisted by some bracing of the body. A seat which provides its maximum comfort when combined with a small amount of muscular discipline will probably be the best choice and it is almost certain to be a more upright than the fully rested position. There is little doubt that the most upright position is the one best suited to take changes in vertical accelerations. Some degree of forward support is offered by the steering wheel but the driver's companion is without such support although the difference does not appear to justify different angles. Even the perfectly shaped seat must allow for some 'shifting around' to relieve fatigue over the areas of support and the cushion should be so shaped and loaded that the backside can be moved forward along it to provide some adjustment to the trunk angle. This should also be sufficient to accommodate the variation between the driving and passenger postures as well as periodic adjustment and differing styles of sitting. In hard braking, the operating effort is conveyed along the leg to the back of the pelvis against the seat. While the rate of deflection is normally required to support the pelvis against rotation, the rate must increase sharply in its 'riding' phase to provide resistance to this brake-operating effort.

In resisting the effects of violent braking and, perhaps in steering, some extra support at the shoulders is essential for the driving posture.

In considering the 'ride' phase of the cushion, a low load/deflection rate is desirable from many points of view. It is, however, highly undesirable for the reason that it increases the relative movement between the squab and the cushion which causes 'shirt-shifting' by the sliding motion between the occupant's body and his clothes. Few cures for this defect appear to have been attempted in seats and those which have been marketed have not, apparently, proved their advantage with much success.

Seat design

Ride Characteristics. Many of the dynamic factors affecting the driving posture have already been dealt with. There is still the fundamental function of the seat as a component of the suspension system of the vehicle to be considered. While the suspension designer may spend much thought and attention to the 'ride' of the vehicle and while extra cost might be added to production costs in order to improve the suspension, much of his good work can be nullified by bouncing seats.

For the best ride and the greatest comfort, it is essential that the seat should be considered as part of the suspension system of the vehicle and its load/deflection characteristics determined along with the characteristics of the rest of the system. Some conflict may arise because the suspension designer does not know enough about seat design and the seat designer does not know enough about the mechanics of suspension. The difficulty is by no means insurmountable. The seat designer can provide the loads to be carried to the suspension designer, who would be able, in return, to provide a graph showing the most desirable load/deflection characteristics. Rigs can then be made to determine the characteristics of experimental seats designed to meet the requirements and, although a lot of trial and error work may be necessary, the ensuing results should justify it handsomely.

It has already been mentioned that a large range of deflection of the cushion is undesirable because it causes large movement between the cushion face and the squab face. It is also undesirable because it causes large relative movement between the occupant and the windows. (The greater feeling of safety and the vastly superior ride obtained directly from the chassis suspension was revealed to the Author when first testing a thinly upholstered form-fitting seat specially built for the use of an engineer for developing suspension systems.)

Coincidence of Trunk Line. A design fault found in almost every type of seat is the existence of two positions of maximum comfort, one for the cushion and one for the squab. Fig. 4 illustrates a section through a typical spring case seat (although the criticism can be levelled against other forms of upholstery too). The point of maximum deflection of the cushion lies midway between the front and back border wires of the spring case. If the occupant sits with his haunch bones directly over this point, the pelvis is 'cradled' by equal pressure on either side. It is, therefore, the most comfortable position on the cushion for resting the pelvis. To obtain support for the back, however, the occupant must sit with his haunch bones far enough behind this line so that his lumbar region is rested against the squab. The cushion, in this position, exerts higher pressure against the back of the pelvis and tends to rotate the pelvis forward away from the squab support. The haunch bones are, as it were, sitting on the side of a hill and descent into the valley is inevitable. Not only is the crutch 'strangled' by one's clothes, but there is a fidgeting as one moves back against the squab again. Gradually one slides down to the middle of the cushion once more, remaining in that position until an aching lumbar causes one to move back against the squab again.

Adverse pelvic roll can also be caused by incorrect load/deflection rates in the ride phase. In pressing oneself into a cushion, one can feel pressure rapidly increasing behind one's haunch bones. This pressure tends to rotate the pelvis forward contrariwise to the need, which is to rotate the pelvis backward against the support provided by the squab.

In riding rough roads the constant jolting causes excessive movement of the middle regions which can become most exhausting. A well-designed seat cradles the pelvis between the pressure forward of the haunch bones and the lumbar support offered by the squab.

Area of Support. The smallest total area of contact is most desirable for the purpose of allowing air to circulate around the occupant's body to dissipate excessive body heat. In providing areas of support, it is obvious that they must taper gently off to give good distribution and to avoid any sharp changes of pressure. Correct attention to support can, therefore, also result in minimum area of contact and maximum 'breathing.'

Variations in Driver Sizes. Mention has already been made of the fact that most of the variations in the sizes of drivers must be accommodated in the seat position and that the variations affect the seat shape only slightly.

It has also been stated above that support under the thighs is needed only to 'cradle' the pelvis and the weight of the flesh under the upper leg. The length of the cushion can be made, therefore, to suit the shortest upper leg and the variations accommodated by overhang.

There are, therefore, only two anatomical dimensions which directly affect seat shape, the distance from the hauchbones to the back of the sacrum and the height from the seat to the lumbar/upper trunk joint.

There is comparatively little flesh behind the sacrum—as one can feel on oneself—and the dimension from the

bottom point of the haunchbones to the back of the sacrum is about 5 in. The variations in this dimension can be accommodated by arranging the point of maximum deflection of the cushion to suit the smallest pelvis in the most upright position. Any increase on this dimension still retains pelvic rotation in the desired direction.

An estimate of the variation to be accommodated in the height of the lumbar/upper trunk joint can be obtained from statistics gathered from a mixed population above the age of eighteen years (full stature is generally reached by this age) for the distance of the elbow from the seat. This dimension has a range from 7½ in to 13 in, i.e. 5½ in variation.

This can be accommodated by extending the height of lumbar support from the top of the sacrum of the shortest person to the lumbar/upper trunk of the longest person and arranging the load/deflection in the reception phase to suit.

Development of standard data

A considerable amount of anthropometrical and other data have been made available by the anatomists working in the field of postural mechanics. Statistical means and ranges of the sizes and weights of various types of populations have been published and can be used by seat designers in establishing for themselves, standard normal and limit profiles around which seats and seat positions can be designed. The lack of recognition of a complex series of centres of gravity for an articulated structure like the human spine (which can be simplified for practical purposes as described in the present article) has led to an unfortunate omission from the seat designer's point of view. The resultant difficulty is not insurmountable and it will be found that separate centres of gravity for the upper and lower halves of the trunk can be arrived at, from the published data by the customary methods used in mechanics for finding centroids.

With these figures it is then possible to determine the component loads and their directions, to make stress diagrams for the required postures, to specify the load/deflection characteristics and, with the use of rigs, to check the seats for conformity.

One of the most valuable uses of load/deflection rigs, beside their use as part of a research method of development, is to help in perpetuating a comfortable seat. In the many styling changes which occur from time to time in car design, the driving position often becomes altered and new seat designs are required. Without working back to fundamental trunk angles, eye levels, load/deflection rates, etc., a new model which is intended to supersede its predecessor with overall advantages, often fails to preserve the seating comfort previously obtained.

For the reasons given, it has not been possible to quote standards and data already gathered and a considerable amount of more work remains to be done before any common standards can be suggested for general use. It is proper that the postural loads required for seat designing are determined scientifically by the anatomists. Not until data of such loads are made available from such fields can standards of their application be presented.

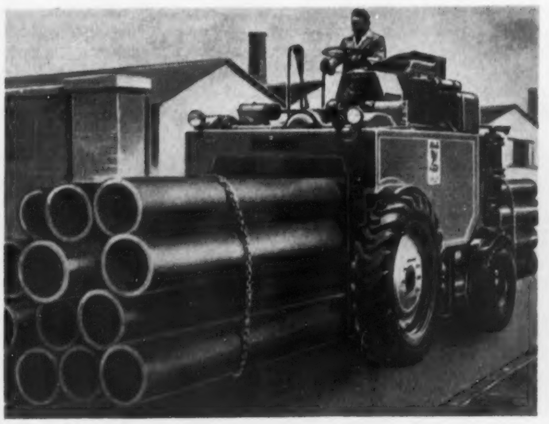
It is hoped, however, that the above discussion might encourage a more critical consideration of seat design and give the hint to anatomists on what part of their work can best be used to practical advantage by the seat designer.

STRADDLE CARRIERS

A New British Development

A NEW development by British Straddle Carrier Co. Ltd., 7, Chesterfield Gardens, London, W.1, is shown in the accompanying illustration. It is known as the "Steelmaster" straddle carrier and is intended for use in the iron and steel and heavy industries. This carrier and the same organization's "Timber Wolf" are the only straddle carriers produced in the sterling area.

The Steelmaster is available with eight different dimensions of load aperture and is capable of handling and transporting loads of more than 40 ft in length and up to 10 tons in weight. It is primarily intended for the conveyance of steel sections, girders, pipes, tubes and bars, but is equally suitable for billets, ingots, steel plates and sheets and various "cube loads" for which special cradles or pallets are available.



Steelmaster straddle carrier

A recent series of tests established that the Steelmaster can pick up a load of 7½ tons of steel, average length 23 ft, transport it over a rough concrete surface, deposit the load and return to the starting point in an average time of 78·6 seconds for the double journey. On this basis it is possible to move more than 300 tons per hour over 220 yards with one straddle carrier operated by one man. Neither tying nor securing of the load was necessary This compares favourably with existing methods of handling, loading and transporting lengthy or bulky loads.

A Perkins P.6 diesel engine is used as the power unit for the Steelmaster straddle carrier. It develops 65 b.h.p. at 200 r.p.m. and provides high power with low fuel consumption and low maintenance costs for a road speed of 25 m.p.h. forward or in reverse. The oversize tyres of the large driving wheels give great tractive effort and allow the carrier to traverse very rough ground without any risk to the load. The inside turning radius is only 9 ft. If desired, radio-telephone equipment can be fitted to provide constant contact between the transport executive and the drivers.

Other features include Girling hydraulic brakes, hypoid differential unit, three speeds forward and reverse, and electric lighting system. There is ready accessibility to all working parts. The driver has complete visibility fore and aft, giving him a clear view of the load when approaching, lifting, during transit and when setting down. The lifting gear is mechanical and is controlled by a single lever. If required, it can be operated while the carrier is in motion.

Hitherto, the U.S.A. has had a virtual monopoly of this type of equipment, and straddle carriers are being used to an increasing extent in American heavy industry. The whole of the 1953 production of Steelmaster straddle carriers has been sold, chiefly for export, despite keen American competition. It is intended to devote part of the output for 1954 to the home market. The Steelmaster should be of great interest to heavy industries.

JET ENGINE PRODUCTION

Some American Developments

ALTHOUGH as yet production engineers in the British automobile industry have not had to consider the problems involved in the production of jet engine components, it is probable they will have to do so in the not too distant future. When this does occur, the production engineer will have to legislate for even closer tolerances than those now used in the automobile industry and at the same time, work will have to be carried out on materials that are much more difficult to machine. Because of this, recent American developments discussed in these notes are not without interest to the automobile engineer although they deal with developments for the production of aircraft engines. The equipment described is used by the Ryan Aeronautical Company in the manufacture of components for the Pratt and Whitney J-57 engine.

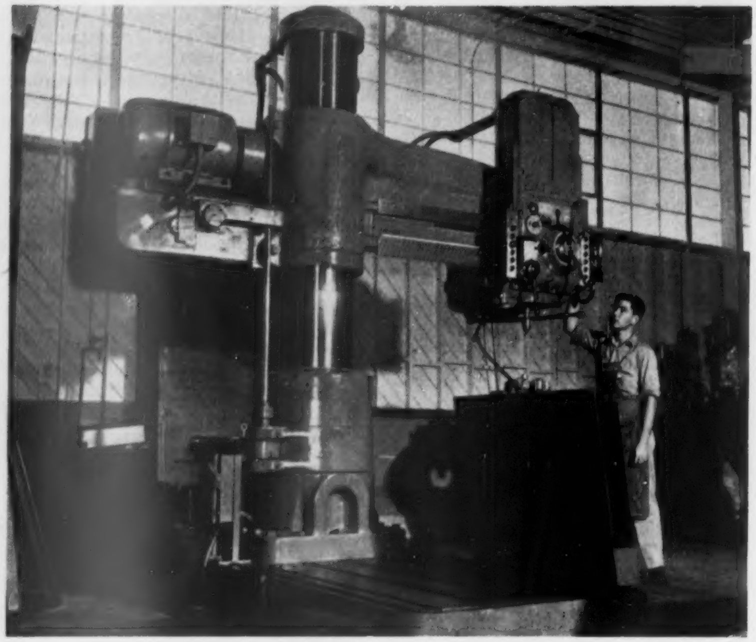


Fig. 1. Carlton radial arm drilling machine

Illustrated in Fig. 1 is a large Carlton radial arm drilling machine that has been specially designed for accurate drilling, reaming and boring operations. The machine has a massive column, three heavy, box section bases and an exceptionally rigid arm and head. An interesting feature of the design is that the main drive gear is located at the lowest point in the head and drives the largest diameter of the spindle. This innovation provides maximum torque transmission from the 15 h.p. motor and eliminates spindle twist and vibration.

Although the Carlton is a very large machine, it is extremely flexible and very smooth in operation. It has 36 spindle speeds in the range 24 to 2,400 r.p.m. For operational convenience, the push button controls are duplicated on the front of the head so that the operator can use either hand. Rapid traverse, and the clamping of the head, arm and column are electrically controlled.

Fig. 2. Lodge and Shipley special "T" lathe

Special tap leads are incorporated in the design so that the machine can be used for precision screw cutting, instead of merely using a tap which is the standard method on radial drilling machines. The three individual bases permit the drill to be used with three large fixtures so that idle time for loading and unloading can be kept as low as possible.

Considerable use is made of a special Cincinnati universal milling machine. This versatile tool embodies two milling heads, each with its own individual motor. It can be used for both horizontal and vertical milling, and because the heads are individually driven a wide variety of work can be handled. A single part can be both horizontally and vertically milled at the same time, or alternatively two parts in separate fixtures can be milled simultaneously.

This machine is specially designed to allow heavy cuts to be taken to close limits of dimensional accuracy. It has a wide range of spindle speeds with a 100:1 ratio and 32 feed rates so that the optimum machining conditions can be used for a wide range of materials. The column, table and carriage are fitted with dial indicators and calibrated standards to allow jig boring operations to be carried out.

An unusual type of lathe is illustrated in Fig. 2. It is the Lodge and Shipley

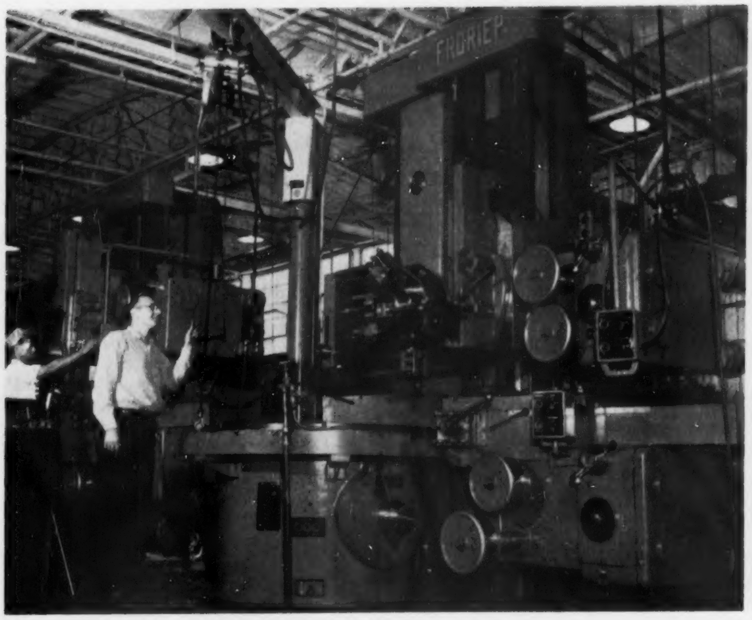


Fig. 3. Froriep vertical turret lathe

'T' lathe that has been developed for facing, turning and boring thin-wall sections of large diameter and short length. It gets its name from the employment of a bed with a carriage-carrying section at right angles to the lathe axis. The machine shown in the illustration has two cutting heads and will accommodate a 60 in diameter component. It occupies less than half the space and costs far less than the smallest conventional type of lathe that will take 60 in sections.

The cutting heads may be operated either independently or simultaneously to perform facing, turning and boring. The large face-plate is arranged to rotate in either direction. Special variable speed driving arrangements that are incorporated in the design are particularly valuable features. They are of a character that allows a constant cutting speed to be applied automatically. This is particularly useful for contouring very wide faces, such as those found in jet engine and turbine parts.

While much of the work that is handled on this machine is of the open ring type that is easily chucked or clamped to the face plate, provision is also made for accommodating work pieces which have shafts or extensions. The headstock spindle has a large diameter hole through which the extension may be inserted. Loading and unloading of large workpieces is facilitated by the unobstructed access to the front of the machine and the extended carriage traverse to the end of the bed.

Fig. 3 shows a German-made Froriep vertical turret lathe, used for operations similar to those performed on the better known Bullard vertical turret lathe. This German machine has a 55 in diameter table that will accommodate parts weighing up to four tons. The main drive is by a 50 h.p. motor and the turning speeds range from 2·8 to 240 r.p.m. Push button controls are employed; they are grouped together on a swivelling pendulum for operational convenience. There is an electro-magnetic disc clutch to ensure fast and safe operation. This machine can be used with equal effect on parts 44 in high and on small components. For example, the machine has been used to machine 20 small fuel tank fittings simultaneously both inside and outside at one set-up.

Because of the need to maintain very close dimensional control, all welded assemblies must be stress relieved by being heated to 1200 deg F. Otherwise, residual stresses set up by the localized welding heat would exert forces tending to distort the contours of the work. A special Knapp furnace, shown in Fig. 4, is used for the stress relief treatment. It is charged by rolling a truck load of parts on tracks extending through the raised door. Heating is effected by burning natural gas in a special com-

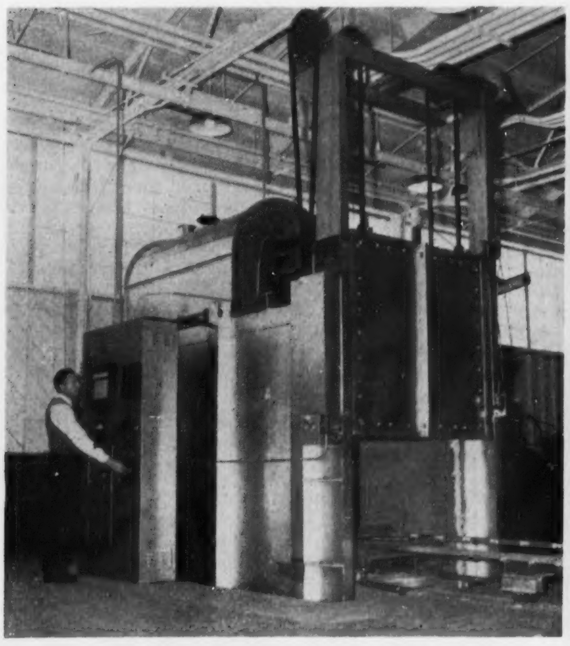


Fig. 4. Knapp stress relieving furnace

Fig. 5. General Electric industrial X-ray unit

bustion chamber on top of the furnace. Powerful blowers thoroughly mix the hot air and circulate it down to the heating compartment. To eliminate any possibility of thermal shock, the heating, soaking and cooling cycles are automatically controlled by electronic means. Since stress relieving temperatures are only moderate, there is no necessity for a controlled furnace atmosphere.

Probably the X-ray inspection requirement is the best indication of the high quality demanded in the production of jet engines. For example, all fusion welding must be 100 per cent X-rayed. This not only ensures that only perfect welds are used in the final product but also acts as an immediate check on production welding. A General Electric OX-250 industrial X-ray unit is used. It is suspended from a jib crane as shown in Fig. 5. The unit can be raised or lowered 16 ft and can be rolled back and forth for a distance of 8 ft on the crane arm. To give the unit maximum mobility, the head is arranged to turn through 360 deg on its vertical axis and 270 deg round the horizontal axis of the unit.

The heart of the unit is a Coolidge line focus tube equipped with a hooded anode that increases the output. A small focal spot is employed; this provides fine detail and high quality radiographs with relatively short focal film distance. The tube is immersed in cooling oil that circulates through cables to a radiator. The maximum output of the tube is 250,000 volts at 10 milliamperes. With this high power adequate definition is obtained through 4½ in of steel.

BALL AND ROLLER BEARING LUBRICATION

NO. 8 in the Vacuum Technical Series of publications is entitled *"Ball and Roller Bearing Lubrication."* It is a most informative work and will be of great interest and use to a very wide circle of engineers, whether their interests are in the design, in the installation, or in the maintenance of machinery incorporating ball or roller bearings. These subjects, despite their great importance in every branch of engineering, are not so thoroughly understood as they ought to be. This booklet should do much to correct that state of affairs.

Primarily, the booklet deals with the operation and proper maintenance of ball and roller bearings from the viewpoint of experience accumulated over many years and from a close study of lubrication requirements. The basic principles of ball and roller bearing operation are briefly described at the outset. Following this, the functions of a lubricant and the factors to be considered in selecting either an oil or a grease as a lubricant are discussed.

Oil lubrication is dealt with in some detail. Advice is given on the selection of a suitable oil and the many different ways of applying the oil are described. Grease lubrication is given similar treatment. Extremely useful information is given on bearing housing design for both oil and grease lubrication. Clear line drawings increase the value of this section.

The next section of the booklet deals with bearing removal, cleaning and replacement. These subjects are dealt with under the following headings:—

Cleaning central lubrication systems
Cleaning bearings and housings.
Cleaning without dismantling.
Dismantling bearings for cleaning.
Cleaning and inspection after dismantling.
Bearing replacement.

Bearing closures serve two main purposes of equal importance; the first is to retain the lubricant in the housing, and the second is to prevent the entry of contaminating influences into the bearing housing and so into the raceways. Many factors determine the type of closure best suited for a specific application and the various forms that may be applied are dealt with in some detail.

The penultimate section deals with failures of ball and roller bearings. This section includes a "trouble tracing" chart listing eight common causes of failure, with remarks on methods of identification, probable causes and remedies. Finally, there are general recommendations of Gargoyle oils and greases for use in lubricating ball and roller bearings. This booklet is available without charge from Vacuum Oil Company Ltd., Caxton House, Tothill Street, London, W.1, or from any of the Company's branches throughout the country.

ALUMINIUM ALLOY FOR BEARINGS

THE traditional bearing material is white metal and for many years this satisfied all engineering requirements, but modern machines demand an increased capacity to support higher bearing loads and increased resistance to failure by fatigue. Investigations of possible new materials, carried out by High Duty Alloys Limited, Slough, Bucks, were centred on the existing range of aluminium alloys, and it was found that some of the standard alloys functioned satisfactorily as bearings when used in conjunction with case hardened shafts and maximum fluid lubrication. These materials failed, however, under conditions of boundary lubrication. Consequently, research work was concentrated on the production of a light alloy that would operate under normal conditions of lubrication and preferably with shafts of lower hardness.

From these investigations, the Hiduminium RR.AC series of alloys was made available, culminating in Hiduminium RR.AC9A, which was found to be the most satisfactory. It did not, however, fill all the requirements of a bearing material. Further work has resulted in the development of Hiduminium 29. This material has been subjected to extensive service trials and has proved suitable for main bearings, big-end bearings and camshaft bushes in internal combustion engines. It has also been used successfully as a bushing in large roll neck rolling mill bearings. Hiduminium 29 can be supplied in various forms, such as chill or gravity die cast bars and tubes, or cast as shells—thereby eliminating excessive machining.

Hiduminium 29 has good machining properties, but it has a certain tendency to "pick-up" and consequently an ample supply of coolant is necessary for all operations. It is essential that the tool should have sharp edges. The recommended rakes and clearances are:

Top rake ...... roughing 25/30 deg; finishing 30/35 deg.
Cutting clearance .. roughing 8 deg; finishing 10 deg.

Present experience suggests that the following operating characteristics should be observed. The maximum allowable steady load should not exceed 4,000 lb/in^2, and the maximum oil reservoir temperature should be 125 deg C. The recommended surface finish for both shaft and bearing is 20 RMS, and the shaft to bearing clearance should be 0·00 in-0·0012 in per inch of shaft diameter.

In most literature, a minimum shaft hardness of 300 Brinell is quoted for aluminium alloy bearings, but tests carried out with Hiduminium 29 have proved that, provided the recommended oil pressure is maintained and an excellent filtration system is employed, shafts with hardness as low as 180 Brinell can be used satisfactorily. Full lubrication is essential and a pressure supply of 20 lb/in^2 is recommended.

In most of the tests a medium oil of S.A.E. 20 grade has been used, but it is expected that lower viscosity oils will be equally successful.

PUBLIC SERVICE VEHICLES

Some New Continental Designs

ALTHOUGH British manufacturers of public service vehicles have, with one notable exception, universally adopted the underfloor location for the engine in their latest types of chassis, Continental manufacturers of similar vehicles show a great diversity of opinion on the subject. Several important Continental organizations now produce underfloor-engined vehicles, but there is also a growing tendency to favour the rear-mounted engine, a trend which is also becoming apparent in Britain. There are several reasons for the failure of the underfloor-engined bus or coach to achieve universal popularity. First, there is the height of the floor, with the consequent increase in the number of steps and the height of the risers necessary to gain entry. Secondly, there is difficulty in obtaining a favourable distribution of weight between the two axles. Finally, because of its exposed position, the engine can be relatively easily damaged on snow-covered roads, and there are cooling problems, since the engine tends to run too cool in winter and to become overheated in summer. Over-cooling is the greater problem, although this is occasionally overcome by fitting close-fitting shrouds around the engine to give better control of the flow of air over the cylinder heads and block, and at the same time protect the engine from mud and dirt thrown up by the front wheels off the road. A third advantage of cowling the engine in this way is that the shrouds tend to act as an acoustic shield, preventing engine noise from reflecting off the road surface and creating a relatively high level of noise in the interior. Against these advantages is the fact that by cowling the engine, some of the accessibility which is obtained by locating under the floor is lost. To overcome the various disadvantages associated with the central underfloor engine position, Continental manufacturers produce vehicles with the engine located horizontally under the front overhang of the chassis, vertically across the front of the chassis, as well as horizontally under the rear overhang, vertically across the rear, and vertically along the rear overhang.

One of the latest additions to the range of rear-engined chassis now available on the Continent is the new Saurer 3HR. This chassis is devised principally as a touring coach with a capacity of 34-38 passengers. For bus work, Saurer offer a range of chassis in which the engine may be located vertically outside the left-hand frame side-member, either in front of or behind the rear axle, or vertically and transversely behind the rear axle. In addition, this company manufactures chassis with the engine located vertically over the front axle, and an integral coach with the engine located vertically along the longitudinal axis of the chassis behind the rear axle. The new model has the engine mounted transversely behind the rear axle, but instead of being vertical, as in the 4HP bus chassis, the engine is tilted backwards through approximately 15 deg; this gives a more compact arrangement than if the engine were vertical, in which case it would not be possible to arrange seats over the engine bay as is the case with the 3HR coach.

The chassis of this design is quite conventional, consisting as it does of channel section longitudinals and cross-members, with the longitudinals parallel up to a point behind the front axle, after which they are swept inwards to run parallel again between the front wheels. The overall length of the complete vehicle is 34 ft 2 in, and the overall width is 7 ft 10½ in; the wheelbase is 17 ft 7½ in, and the complete vehicle in prototype form, with an all-metal body, weighed 7 tons 18 cwt. Chassis weight is 4 tons 18 cwt. A Saurer CT2D-Lm six-cylinder direct injection engine is used. It is mounted on rubber at three points. The bore is 115 mm and the stroke 140 mm, and the capacity is 8·72 litres. A Saurer injection pump is mounted on the outside of the engine and is driven by a shaft from the single-cylinder compressor which is mounted at the front of the engine and is driven by the timing chain. On the other side of the engine is a Saurer scroll-type supercharger. This is a new design, and is driven by belts from the nose on the crankshaft at approximately 3·4 times crankshaft speed. The total power output of the engine is 170 b.h.p. at 2,000 r.p.m. The normally aspirated version of this engine produces 125 b.h.p. at the same engine speed.

From the rear of the engine (the left hand side in this installation) the drive passes through a shaft, flexibly jointed to the flywheel at one end and to a bevel gear at the other, to the two-plate clutch, hydraulically operated through a servo-motor. From the clutch the drive passes through a Saurer four-speed gearbox and two-speed auxiliary box to a hypoid rear axle. The axle ratio is 5·43:1. Either 9·00 by 20 or

The rear seats on the Saurer 3HR coach are well back; all the seats are adjustable for rake

10·00 by 20 tyres (twin rears) may be fitted. The brakes are hydraulically operated through a servo system, and an exhaust obturator is a standard fitting. Semi-elliptic springs controlled by telescopic shock absorbers are fitted all round. The electrical system is 24 volt.

The advantages of tilting the engine backwards are seen in the interior of the coach. Only a slight slope makes it possible to take the floor over the top of the cylinder head, without seriously affecting engine accessibility, which is generally of a high order. At the same time, reasonable headroom can be maintained throughout the length of the vehicle. Unlike earlier Saurer rear-engined coaches, air for cooling and for the engine is not taken in through a duct in the roof. A large grille on the right hand side of the body covers the radiator, but air for cooling and combustion is drawn in from the front of, and underneath, the engine bay. A full-length duct in the roof of the coach is arranged to provide adequate interior ventilation. Air is taken in through a large grille in the front, and emerges through a series of outlets in the interior. At the rear of the duct there is a controllable extractor vent to remove vitiated air. As the entire roof on each side of the duct is glazed, the problem of ventilation is a serious one. For heating the interior, an Eberspächer diesel oil-burning installation is provided. Space for luggage is provided behind the rearmost seats inside the body, and in large lockers under the floor, reached through external hatches. The spare wheel is concealed under the front overhang.

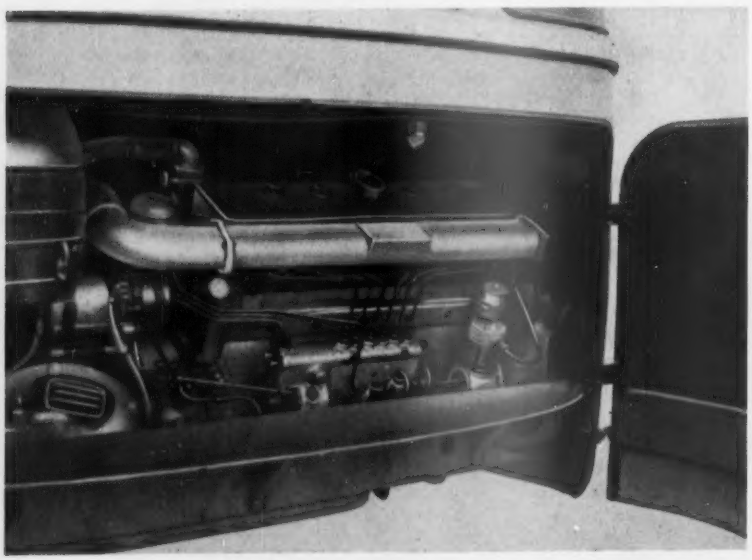
A marked contrast with the Saurer is provided by the Ikarus 55, a new design of integral bus or coach developed by a Hungarian concern, represented by Mogurt, the Hungarian Motor Vehicle Trading Co., of Budapest. Like the Saurer, the Ikarus 55 has a rear-mounted six-cylinder engine, but in this case, the engine is located vertically and longitudinally in the rear overhang. An indirect injection Csepel 613 engine is fitted. It has a bore of 110 mm, a stroke of 140 mm, and a swept volume of 7,983 c.c. At 2,200 r.p.m. the engine produces 125 b.h.p., while the maximum torque output is 352 lb-ft at 1,600 r.p.m. The wheelbase is 18 ft. $2\frac{1}{2}$ in, the front track 6 ft 1 in, and the rear track 5 ft $11\frac{1}{2}$ in in the centre of the twin rear wheels. The overall length is 37 ft 6 in and the vehicle is equipped with 12·00 by 22 tyres. On left lock, the turning circle is 55 ft 9 in, and on right lock 65 ft 7 in.

The engine is mounted in unit with the gearbox on a sub-frame that is bolted to longitudinal members of the structure at the rear. By this means the engine-gearbox unit can be withdrawn through the rear panelling, which is hinged to facilitate maintenance and overhaul. The clutch is a dry single-plate design, operated by air-pressure. The control valve for the clutch servo motor is bolted to the underside of the plate on which the clutch pedal is hinged. A five-speed gearbox is employed. It is operated manually through a series of rods and linkages running under the floor of the vehicle, through a duct which also contains the brake pipe lines. The gear lever is mounted on the steering column. From the gearbox, two short shafts, with Mechanics 7C universal joints transmit the drive to the double-reduction rear axle. The differential has a spiral bevel pinion and crown-wheel and the overall reduction provided by the rear axle is 6·24 : 1. Semi-elliptic springs are used on both front and rear axles, the front springs being conventionally shackled at both ends, whereas the rear ends of the rear springs have sliding shackles. Globoid worm and double-roller steering, with a ratio of 29 : 1, is used.

A twin-cylinder compressor mounted on the left of the engine and driven by belts from the front of the crankshaft provides power for the air-pressure braking system. In addition to the normal air reservoirs, however, there is a third unit fed from one of the main reservoirs. This provides compressed air for a servo-operated hand-brake, which is fitted in addition to the normal mechanical hand-brake, and which operates on the rear wheels only. Individual cylinders are provided for each wheel, with an extra pair, actuated by the emergency hand-brake, for the rear wheel drums.

The cooling system has been the subject of careful investigation, with the object of achieving optimum cooling, reducing noise from the radiator fan, and of keeping the engine bay and combustion air as clean as possible, experience having shown that rear-engined vehicles draw dust and dirt in from the region of low pressure air immediately behind the vehicle. For this reason the engine bay is carefully sealed, as well as being sound-proofed, and air enters the bay only through a duct connected with two adjustable vents in the roof. The duct carries air down to the radiator located across the right hand side of the engine bay, and in line with the rearmost cylinder. A six-bladed fan behind the radiator is driven by a belt passing over a pulley mounted immediately in front of the clutch housing. The pulley itself is driven by an external shaft running along the engine in line with the cylinder-block sump joint, from a second pulley, belt driven off the nose of the crankshaft. The fan-driving belt is triangulated over the pulley, fan and a dynamo mounted behind the radiator. Hinged panelling enclosing the engine bay at the rear has a wire-mesh grille, through which air passes to the atmosphere propelled by the radiator fan. Electrically welded pressed steel channel and folded box-section members comprise the structure of the Ikarus. The cross-members and longitudinals in the underframe are mainly channel sections, but the pillars, crib, waist and cant rails are box sections. The roof is framed in top-hat section formers and stringers. Between the waist and crib rails, and around the wheel arches, diagonal members are introduced to increase longitudinal stability. In the coach version of the design, a central entrance, 4 ft 2 in wide, enclosed by double, hinged doors, is provided. The Ikarus 66, a bus version of the design, has a single, jack-knife front entrance, and a double jack-knife central exit.

The Ikarus 55 carries 44 passengers, while over 70 passengers may be accommodated in the bus, which is equipped with 32 seats. Both the bus and the coach are panelled in aluminium internally and externally, while the coach has luggage lockers under the floor, reached through external doors. These lockers have a capacity of

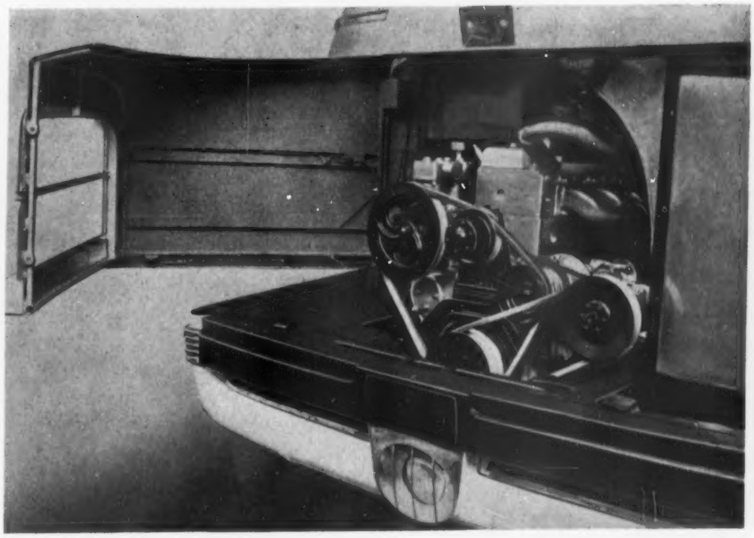

In the Saurer 3HR coach, a compact installation is obtained by tilting the engine to the rear

106 ft^3, additional lockers being provided for the spare wheel and tools. The batteries are located on the left of the engine bay, with the fuel tank occupying the equivalent position on the other side of the bay under the radiator. Seven lamps are concealed under the front bumper assembly—two headlamps, two fog lamps, two outward-facing cornering lamps, and a spot light.

Among the underfloor-engined bus chassis manufactured on the Continent, two of Swiss manufacture command particular respect. One is the F.B.W., manufactured by Franz Brozincevic and Co., Wetzikon, Zürich, and the other is the Mowag, produced by Motorwagenfabrik A.G. Kreuzlingen. The F.B.W. has a six-cylinder horizontal engine, and the Mowag an air-cooled, eight-cylinder horizontally-opposed unit.

The F.B.W. has been subjected to careful development over several years, and represents a fine example of classic design and attention to detail. Channel section side-members are used in the chassis. They are parallel from the front cross-member to a point in line with centre of the gearbox, after which they taper slightly, to run parallel again to the rearmost cross-member. Side- and cross-members are assembled by electric welding. The chassis is available with wheelbases of 17 ft 8½ in and 18 ft 4½ in. In either case the front and rear overhangs are 6 ft 10½ in and 10 ft 6 in respectively; the overall length of the long wheelbase model is 36 ft and the chassis weighs 5 tons 8 cwt. The smaller model weighs 5 tons 4 cwt in chassis form. Gross vehicle weight in either case may be 14 tons 15 cwt.

Similarity with British underfloor-engined vehicle design is seen in the location of the engine with its centre line almost one-third of the distance between the front and rear cross-members. Unlike similar British vehicles, however, the engine is mounted at a slight angle to the longitudinal axis of the chassis, thus offering improved accessibility to the oil cooler mounted in front of the cylinder block, and to the compressor and injection pump located on top of the cylinder block. The cylinder heads lie to the left of the crankshaft. The pump is mounted with its centre coinciding with the transverse centre line of the cylinder block so that the leads to the injectors are relatively short and symmetrically disposed. Two rubber mountings, one each side of the clutch housing, support the end at the rear, with a single rubber mounting at the front.

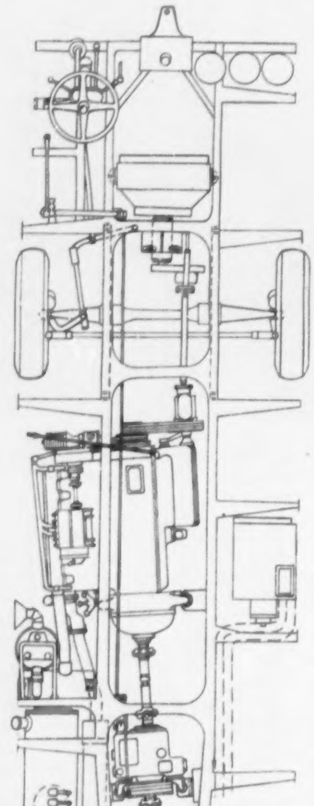


Good accessibility to the engine and auxiliaries is obtained in the Ikarus 55 by the provision of large hinged doors extending round the sides and rear of the engine compartment

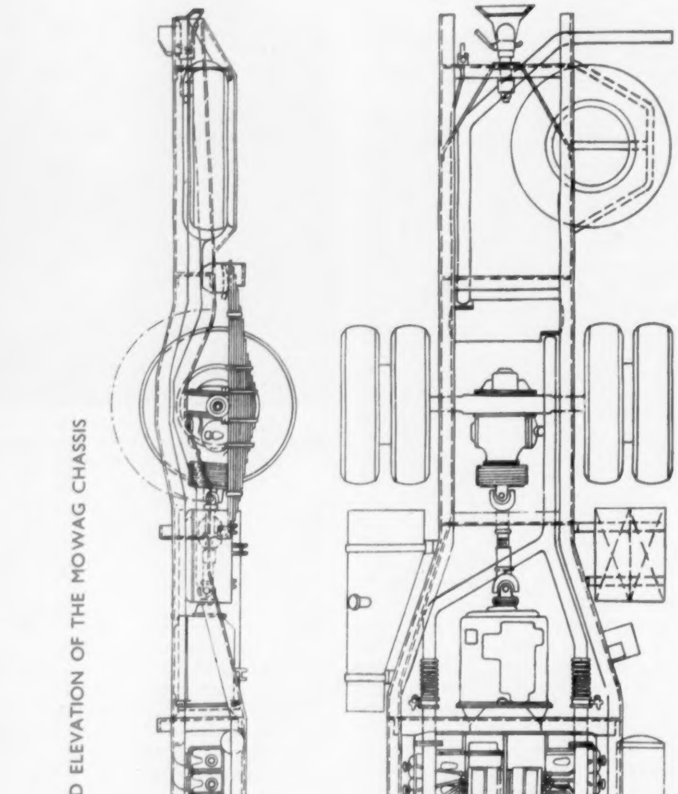
The six-cylinder engine has a bore of 125 mm and a stroke of 150 mm. Its capacity is 11·02 litres. The two cylinder heads carry three valves per cylinder, there being two masked inlet valves and a single exhaust valve. Fuel is injected direct into a combustion chamber in the head of the light alloy pistons. To reduce the length of the push rods, the camshaft is located high up the walls of the cylinder. The compression ratio is 16·3 : 1. Push fit wet cylinder liners are used, and the top piston ring is chromium plated. A dry sump lubrication system is employed. Maximum power output for bus operation is 145 b.h.p. at 1,800 r.p.m. and the maximum torque reaches the high figure of 470 lb-ft between 1,000-1,100 r.p.m. The torque output between 800 - 1,400 r.p.m. exceeds 440 lb-ft.

In unit with the engine is a Vulcan-Sinclair fluid coupling. From this the drive passes through a short shaft with a universal joint at each end and a sliding joint in the centre to a separately mounted, air-operated Wilson

Adjustable air intakes in the roof of the Ikarus 55 ensure that adequate quantities of clean air reach the engine and the radiator



PLAN VIEW OF THE F.B.W. CHASSIS



PLAN AND ELEVATION OF THE MOWAG CHASSIS

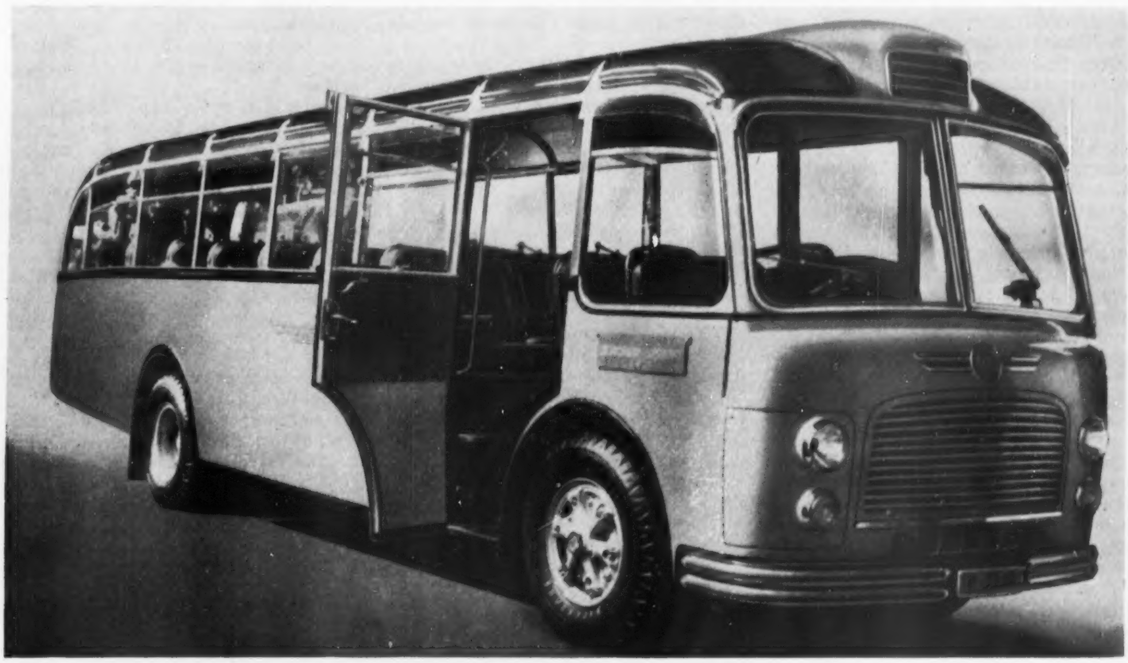
epicyclic four-speed gearbox. Two-pedal control is arranged in conjunction with this gearbox. In unit with the gear-selector under the steering wheel is a series of valves controlling the flow of compressed air to cylinders in the lower half of the gearbox, which engage the brake bands of the lower three speeds through bus-bars. The direct drive multi-plate clutch is also operated by compressed air. An unusual feature of this installation, and of the underfloor-engine mounting, is that the same units are employed for goods vehicles of the company's manufacture.

Buses and goods vehicles with underfloor engines are also built by the Mowag organization. The engines are manufactured by the Société Suisse pour la Construction de Locomotives et de Machines, of Winterthur. All are horizontally opposed, air-cooled units, and are available in four, six, or eight cylinder versions. Only the largest unit is employed for the bus models, but a 4-ton goods vehicle is made with the four-cylinder unit, and a 6- and a 9-ton model are produced with the six- and eight-cylinder units. These engines have a bore and stroke of 110 and 140 mm respectively, the capacities of the various units being 5·33 litres, 7·995 litres and 10·66 litres. The largest unit produces 165 b.h.p. at 2,200 r.p.m. and the other two, 125 b.h.p., and 85 b.h.p. respectively. The maximum torque output of the eight-cylinder engine is 434 lb-ft at 1,400 r.p.m.

Seven lamps are concealed under the massive bumper assembly of the Ikarus coach

This unit is fitted in a patented chassis, in which the engine is mounted slightly forward of the centre of the wheelbase. The upper flanges of the chassis side-members are level throughout their length apart from a slight curve over the rear axle. Angle-section side-members run parallel from the front cross-member to behind the front axle, after which they are sharply swept out around the engine bay. They are swept in again behind the gearbox and run parallel to the rear of the chassis. The engine is cradled between this upper pair of side-members, which are level from front to rear, and a lower pair, which are welded to the upper pair to form a channel section where they are parallel, and then are cranked downwards to pass under the upper pair, outside the cylinder heads. Two deep cross-members and gusset plates in the angle between the level upper side-members and the cranked lower side-members impart torsional rigidity to the frame. Additional rigidity is conferred by attaching the floor directly to the upper flanges of the side-members. The engine is surrounded by a light alloy cowling.

In unit with the engine is a single dry-plate clutch and a Z.F. six-speed gearbox, controlled from a lever on the steering column. The clutch is



The F.B.W. underfloor engined coach has a relatively short front overhang. A grille is provided in the front panelling, because the radiator is ahead of the front axle

hydraulically operated. A short shaft links the gearbox with the double-reduction rear axle, in which the overall reduction for a bus with a total capacity of 120 passengers is 8·6:1. Normally, the tyres are 10·00 by 20, 16-ply, with twin rear wheels. Ross steering gear is fitted as standard. The chassis of a maximum capacity city service bus weighs approximately 6 tons. Despite the relatively large engine, the laden floor height is only 2 ft 11 in, the laden height of the upper flanges of the chassis side-members from the floor being approximately 2 ft 7 in. For a 36 ft long bus, a wheelbase of 18 ft 4 in is employed, the front track being 6 ft 6 in and the rear track 5 ft 8 in.

Although traps in the floor of the body permit access to the two fuel injection pumps in this design, maintenance can normally be carried out only from underneath. On the other hand, the width between the side-members of the chassis surrounding the engine is 4 ft 11 in, so that lowering the engine out of the frame is not difficult.

Compactness is a noteworthy feature of the F.B.W. 11 litre, horizontally opposed, six-cylinder engine for passenger and goods vehicles

TRENDS IN ENGINE LUBRICATION

Notes on Recent and Probable Future Developments

TWO topics immediately suggest themselves under this head: the wider application of relatively high additive level oils and "ultra high viscosity oils." In recalling the steadily increasing use of additive oils, and particularly those embodying a "detergency/dispersancy" function, it may be of interest to note that in the United States of America the oils that are first recommendations even for private cars of manufacturers such as the General Motors Corporation are full "heavy duty" type, and that such oils are now generally used in the U.S.A. when here a "premium" type would be used. Far from there being any misgivings regarding "detergency" *vis-a-vis* petrol engines, the U.S. market is glad to have it and make use of it, particularly when this makes a critical difference to the efficient running of engines with hydraulic tappet adjustment. From the point of view of the engine designer there is every reason for taking advantage of this property in an oil. There may be valid reasons for a similar trend to emerge in this country.

With regard to additive oils for C.I. engines, the author would suggest that there is a most powerful case for the general use of oils of at least DEF/2101 or MIL-O-2104 level. While these cater adequately for the needs of most British units, a demand for oils of yet higher additive concentration is developing. For example, a recently developed two-stroke engine requires a 2104 B Supplemental List 1 oil; in this case, the demand is dictated by a desire to ensure maximum freedom from "port blockage." It may also be that with the natural tendency to look for increasing power outputs from four-stroke units from a given swept volume, there will be further demands for oils of this type.

At first sight, ultra-high viscosity index oils appear to have great attractiveness, but considerable field investigation will be necessary before the usefulness of such oils to the fleet operator can be validly assessed. The allure of oils that can be described as S.A.E. 5W/20 or S.A.E.10W/30 as an aid to ease of starting under very cold conditions is one thing; their potentialities for fleet operators, many of whom are already successfully using S.A.E.5W or 10W oils, is another.

The normal upper limit of engine lubricant base oils is 100 viscosity index, with a limited availability of 105 V.I. oil obtained by special refining technique, and primarily for aviation purposes. V.I. depends upon the origin of the crude oil and the method of refining, and apart from the fact that the lower V.I. oils have to date been somewhat less costly than high V.I. oils, they have enjoyed, and may continue to enjoy, considerable support for some applications, because of their own particular individual properties, which can outweigh their steep "curve." The first use of V.I. "improvers" has been to offset this, a practice not generally considered with favour in this country. Indeed, many purchasing specifications definitely preclude their use. There has been sound reason for this in that the long term effectiveness of the V.I. "improver" has been suspect in regard to breakdown under conditions of severe rates of shear. However, V.I. "improvers" of reliable behaviour are available, and oils containing them have been introduced to the U.S.A. on a limited scale in the lower viscosity grades.

S.A.E.5/20 or S.A.E.10/30 oils might have some appeal to private motorists and fleet operators who for various reasons require an oil of conventional viscosity at normal sump temperatures, but would like easier starting and some possible fuel economy at low temperatures. They might also be attractive to the engine builder who, while attracted to the advantages of low viscosity oils, feels prevented from generally endorsing them because he is bound to allow for engines that may be employed in unsuitable mechanical or operating conditions.

Reference, by way of simple illustration, to the two above-mentioned types, is not meant to connote any implied rigidity in approach, for a wide range of variations is possible—for example, oils of the S.A.E.5W low temperature requirement, but behaving at normal engine temperature as 100 V.I. S.A.E. 10W oils could be of interest in the search for maximum fuel economy. An S.A.E. 10W/20 blend might fill a useful niche, making some fuel economy safely possible in conditions that at present preclude the use of low viscosity oils.

Abridged extract from a paper by F. W. Lawrence, A.M.I.Mech.E., Shell-Mex and B.P. Ltd.

GUDGEON PIN DESIGN

Some Aspects of the Design of One of the Most Severely Loaded Components in Reciprocating Type Internal Combustion Engines

SURPRISINGLY, there are still some, even among experienced designers, who think that the gudgeon pin is a relatively unimportant feature that can be incorporated almost as an afterthought in an engine design. In fact, it is one of the most highly loaded components of the engine and has to operate under exceptionally severe conditions. Now that many manufacturers who hitherto have made only petrol engines are turning their attention to small diesel units, it is increasingly important that the factors relevant to gudgeon pin design should be widely appreciated.

Gudgeon pins operate at temperatures of the order of 150 deg C and are heavily loaded by the combustion pressures and by inertia. Furthermore, not only are they in many instances sparsely lubricated, but the relative motion between the pin and its bearings is of an oscillatory type. In fact, it is probably true that during the development of prototype engines more troubles are experienced with gudgeon pins than with pistons.

Materials

Different materials are used for different applications. In general, En 32 gudgeon pins are employed for petrol engines in private cars and light commercial vehicles. For more highly rated petrol engines and for diesels, En 36V or En 354 are favoured by most manufacturers. In very highly rated engines for racing and for some aircraft, the high tensile steel En 39 is employed, because thinner sections can be used and this reduces the reciprocating weight. However, when such sections are called for, a careful check must be made to ensure that the deformation under load will not be greater than can safely be permitted.

A number of proprietary steels are also employed but generally their composition approximates to one of the British Standard specifications already mentioned. Nodular cast irons are under consideration, with a view to using them for gudgeon pins. Another development that shows promise is the chromium plating of pins to reduce wear.

The heat-treatments generally recommended for the B.S. steels are as follows. All of them are carburized at 880-930 deg C. Subsequent to the carburizing process, it is necessary to refine En 32 at 870-900 deg C and oil quench. Then the pins have to be heated to 760-780 deg C and water quenched. If En 354 is used, it is refined at 850-880 deg C subsequent to the carburizing treatment and then oil quenched. Next it is hardened at 780-820 deg C and oil quenched. The treatment for En 36 is the same as for En 354 except that it is hardened at 760-780 deg C. If En 39 is employed, it is first carburized at 880-930 deg C, as are the other materials. Then it is refined at 850-880 deg C and cooled in air or oil. Subsequently, the components are hardened by quenching in oil from a temperature of 760-780 deg C and then tempered at a temperature not exceeding 200 deg C.

It is usual, of course, to include test pieces in the carburizing packs. After removal from the furnace, these test pieces are fractured to check case depth. Gas carburizing is sometimes employed instead of the pack hardening process. The type of oil used for quenching is not critical.

Induction hardening is sometimes applied to gudgeon pins. Nevertheless, some authorities consider this process to be unsatisfactory for the manufacture of this component, because hardening cracks may develop on the surface. Although the cracks are minute, they cannot be tolerated in gudgeon pins because of the fatigue loading to which these components are subjected. The reason why these cracks develop is that the heat cycle is so short that a uniform temperature is not obtained throughout the section of the pin, so thermal stresses are induced. Efforts are now being directed towards overcoming this trouble by passing a small water-cooled induction coil through the bore of the pin simultaneously with the application of the external coil, but many problems remain to be solved before this development can be regarded as a sound practical proposition.

Extruded or drawn tube is sometimes used for gudgeon pins because it is much cheaper than machining from bar. However, material in this form is not generally favoured, because seams or inclusions may extend along the total length of the tube, and other defects such as those caused by rough or worn dies may be present. These defects can initiate cracks which, under fatigue loading, may spread and quickly cause failure. When black bar is used, it is advisable to machine a 1 in diameter pin from 1¼ in diameter bar to ensure that all rolling seams and inclusions are removed.

After rough turning, the bar is parted off in appropriate lengths and drilled axially. The drilling operation is performed to reduce the cross section of the material to a thickness almost equal to its finished size and, by so doing, to save time in the subsequent heat-treatment. When the components have been carburized and annealed, they are again drilled, this time to remove the case from the bore. It is possible, by plugging the ends of the pin with special compounds or by copper plating the bore, to avoid having to do the second drilling operation. However, plugging is not always entirely reliable because of the tendency of the compound to shrink, and copper plating is expensive. The reason why it is considered desirable to remove the hard case from the bore is that hardening cracks are likely to be formed in it and these might extend and lead to fatigue failures. An important feature of gudgeon pin manufacture is the production of a good surface finish in the bore of the pin. This is necessary because a rough machined finish such as that shown in the illustration of a failed pin introduces notches, or stress raisers, which lead to fatigue failures. Good service has been obtained from pins with bores ground to give an internal finish of 60-70 micro-in. Some manufacturers finish the bore by broaching instead of grinding, but this process may set up minor surface flaws and

Helical grooves are incorporated in the bushes in the gudgeon pin bosses of this two-stroke engine piston, and the ends of the pin are sealed by the plain discs retained by wire circlips

stresses and for this reason does not find favour with all.

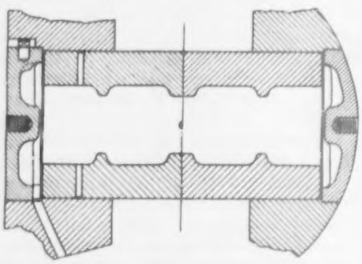
The final operation is the machine lapping of the outer periphery of the pin to obtain a surface finish of 3-5 micro-in. Some relaxation of this standard is permissible on components of larger size, that is on pins of 40-100 mm diameter. It is advisable that the finish should be no smoother than 3 micro-in; if it is, there is a tendency for the lubricating oil not to wet the surface.

Fully floating gudgeon pins

In this country, the majority of manufacturers favour the fully floating pin arrangement. This layout is particularly suitable for engines with small connecting rod centre-to-centre length: stroke, and large bore: stroke ratios, because the large angular movement of the connecting rod obtained with these proportions tends to impose relatively heavy thrust loads on the piston skirts. These thrust loads are of an alternating nature and cause the piston to pant, or deflect alternately in and out, and it is, therefore, essential that relative movement can take place freely between the pin and bearings in the bosses.

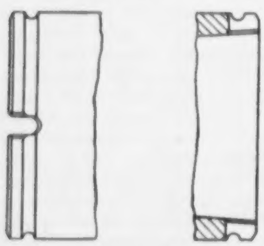
If movement cannot take place in this way, there is a tendency for the piston bosses to creep outwards along the pin. This results in a four-point seizure, that is, seizure of the piston skirt in an area on each side of each end of the gudgeon pin. When aluminium alloy pistons are employed, there is, of course, a tendency for the clearance between the pin and the bore of the bosses to increase slightly as the engine warms up.

Pins in Y-alloy or in low-expansion aluminium alloy pistons used under normal operating conditions should, when lubricated, be a hand push fit in the cross bores of the piston bosses. They must be carefully fitted in the pistons partly by selection and partly by burnishing the bores of the cross holes where necessary. In these circumstances, it is necessary to warm the piston slightly to remove the pin by hand. This is because, with such a close fit, metal-to-metal contact takes place between the pin and the bosses after the oil film has had time to flow from between the asperities of surface roughness. The reason why this care is necessary in the assembly of the pin and piston is that when the piston is at its operating temperature the clearance between the two components must not be too large, otherwise hammering and subsequent failure will take place. On the other hand, it must be large enough to contain an adequate oil film.



The end pads are a snug fit in this piston, as can be seen from the vertical section to the left of the centre line and the plan section to the right. They are machined *in situ* to conform to the cylinder bore. The ribs in the bore of the pin retain a small quantity of oil for lubrication when starting and in case of temporary failure of the supply

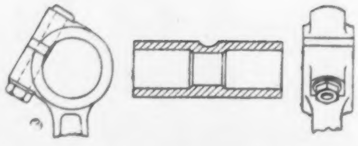
When cast iron pistons are used in engines in which the gudgeon pins are free to rotate in the small ends of the connecting rods, rotation of the pins relative to the piston bosses is, in most instances, prevented by set screws or other devices. If the pins are anchored in the small ends, the most common practice is for the piston bosses to be bushed. In bushed or unbushed cast iron pistons, the pins should be a running fit in their bosses.



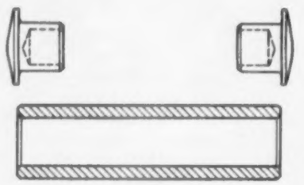
An elevation and a plan section of the ends of a gudgeon pin that is located by circlips on its ends. The slots are incorporated to facilitate removal of the circlips

Axial location

The practice of using spigoted-in end pads of a soft material, such as aluminium, brass or bronze, to prevent the ends of the pins from scoring the cylinder walls, is dying out. However, it is still adopted in engines of very small bore and, in some cases, where the big ends of the connecting rods are too large to pass through the bores. Units of this type are dismantled in the following order: the connecting rod big end bearing cap is removed, the piston pushed up through the bore and the gudgeon pin removed, and then the connecting rod is withdrawn downwards. When end pads are used, this sequence of operations is effected more easily than when it is necessary to remove circlips.



If the pin is clamped in the small end, it should be of thick section at the centre to avoid distortion

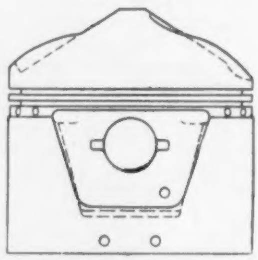


When end pads of this type are employed, they should be a good fit in the bore of the pin, which should be free from tool marks

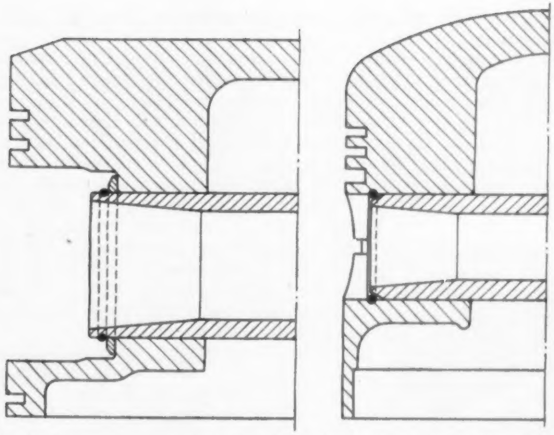
It is good practice to specify a slight interference fit between the shank of the cap and the bore of the pin, otherwise hammering tends to take place under inertia loading and the pad will eventually disintegrate. On the other hand, if the interference is too great, differential rates of expansion between the plug and the pin will cause the counterbored shank of the plug to collapse. Tool marks should be removed from the bore of the pin because they may tend to shear off some of the surface of the relatively soft material of the shank of the pad when it is being fitted and the specified interference fit will be lost. Moreover, pads tend to bear eccentrically, to a small extent, on the cylinder bores and this causes them to rotate. In these circumstances, further damage may be caused by tool marks in the pin.

End pads are also frequently used on two-stroke engines in which the ports are incorporated in the cylinder walls. In this application, the caps are employed to seal the cross holes in the piston so that they do not interfere in any way with the action of the porting. Engines of this type often have pressure lubricated small ends and the oil is passed through a radial hole to the centre of the pin and thence along the bore to the cross holes in the piston. In such applications, the pads are used to retain the lubricant and to avoid oil control difficulties, since oil control rings cannot always be fitted above the gudgeon pins in this type of engine. In some designs, the end pads are retained by a tie-bolt through the gudgeon pin bore. The head of the bolt and the nut are, of course, countersunk in the pads. In other cases, the pads are retained by circlips in the shouldered ends of the cross holes.

In four-stroke engines, axial location of the gudgeon pins is commonly effected by circlips in grooves in the outer ends of the bores in the piston bosses. Circlips of the Seeger type are usually employed for pins of diameters up to 3½-4 in and, provided they are correctly fitted, there is little risk of failure. However, a point that is frequently overlooked is that recommendations should be made in servicing manuals to the effect that the old circlip should not be replaced after it has been removed for servicing. It is



When wire circlips without turned-in ends are fitted in the piston bosses, slots must also be incorporated so that they can be removed



Left: An aero engine piston in which the gudgeon pin is retained by plain washers, and circlips in grooves round its ends. Right: When plain wire circlips are fitted in grooves in the piston bosses, the ends of the gudgeon pin are often chamfered so that they help to retain the circlips in their grooves

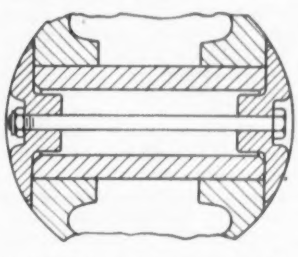
necessary to fit a new circlip because there is a tendency for the old one to lose some of its elastic properties during service and for it to be overstrained on removal.

Fairly close tolerances should be maintained on the overall length between the circlip grooves and on the overall length of the gudgeon pin. The axial float should be as small as possible; if it is too great, pounding may take place and failure of the land outboard of the circlip may result. When wire type circlips are employed, this pounding may cause them to be dislodged.

Wire type circlips are favoured for racing engines and other units in which the inertia loading is high. They are lighter than Seeger circlips, but are not so easy to remove. In some instances, the ends of the circlip are turned inwards to facilitate removal, but with this arrangement it is necessary to use square-ended gudgeon pins which may tend to dislodge the circlips from their grooves. In high speed engines it is more usual to use circlips without turned-in ends, and to facilitate their removal by providing two diametrically opposed slots in the ends of the bores. With this arrangement, the ends of the gudgeon pin should be chamfered so that, if the pin moves axially, they tend to force the wire into the groove, instead of to gouge it out. In these circumstances, the maintenance of close tolerances on bore and pin lengths is most important. The reason why wire type circlips are not more commonly used in engines designed for moderate speeds is, of course, that this arrangement is more expensive than that of Seeger type circlips.

In aircraft engines, axial location is sometimes effected by means of wire type circlips sprung into grooves round the ends of the pin. In this case, conical washers are sometimes fitted, with their concave faces outwards, inboard of the circlips so that axial movement of the pin relative to the piston tends to force the circlips into their grooves. Slots to facilitate the removal of the circlip are machined in the ends of the pin. The main disadvantage of this method is that it is expensive. However, it is used in aircraft engines because of the need to reduce weight to an absolute minimum.

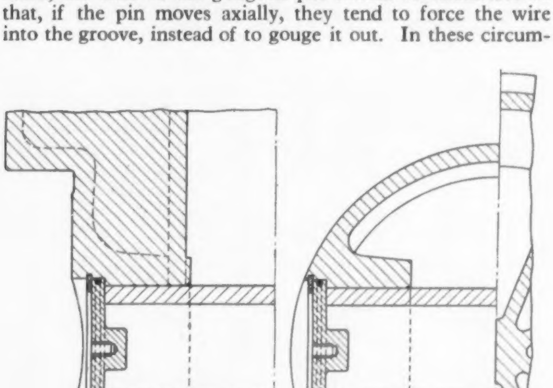
The cylinder bore of this type of unit is almost invariably large and the piston short. In these circumstances, the most economical way, so far as weight is concerned, of supporting the loads on the piston crown is to position the bosses and struts some distance inboard of the piston skirt. This shortens the span of the crown between the struts, thus reducing the bending loads on it and therefore the weight of metal necessary to carry those loads. The ends of the pin can, therefore, be made to project outboard of the bosses to carry the circlips and yet still be clear of the cylinder bores. To mount the circlips in grooves in the cross holes would involve increasing the length of the bosses and add to the reciprocating weight.



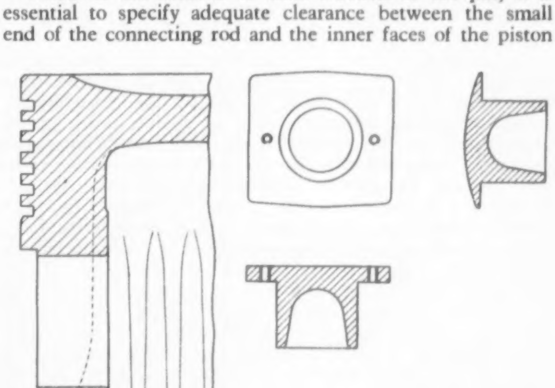
A tie-bolt is sometimes employed to retain the end pads

In some engines, mainly of low rating, axial location is effected by clamping the gudgeon pin in the small end of the connecting rod. A groove is usually machined chord-wise in the periphery of the pin, and the clamping bolt registers in it to provide positive axial location. To counteract the tendency for the gudgeon pin to be distorted under the clamping action, the bore is stepped to give a larger cross section at the centre than at the ends.

Whatever the method of axial location of the pin, it is essential to specify adequate clearance between the small end of the connecting rod and the inner faces of the piston



In this compressor piston, each end of the gudgeon pin is sealed by a circular plate, with a compression ring round its periphery, and the whole assembly is retained by a Seeger circlip. The tapped hole in the plate is for an extractor bolt



A piston, gudgeon pin and end pads of a diesel engine of about $10\frac{1}{4}$ in bore. The step at each end of the bore of the pin is incorporated because close tolerances are called for there to accommodate the shanks of the end pads

bosses. This clearance should be a minimum of $\frac{1}{32}$ in per side in the smaller engines used in private cars, and $\frac{1}{16}$ in per side in engines of up to 150 mm bore used in commercial vehicles. If the clearance is not big enough, the small end may tap against the piston boss and give rise to a noise that will probably be associated with piston slap. On the other hand, the clearance must not be too large otherwise the bending loads imposed on the gudgeon pin will be severe. The tendency for the connecting rod to move sideways and cause the tapping noise is probably due to crankshaft whip.

Bearings and lubrication

In well designed engines, the temperature in the bores of the cross-holes, due to the conduction of heat from the crown, is about 125-150 deg C. This is true both of petrol and diesel engines; for, although the crown temperatures in diesel units are higher than in petrol ones, the distance between the crown and the boss is also larger. The temperature rise due to friction between the pin and the bore should be approximately a further 20-30 deg C; between these limits, it will vary with the loading and method of lubrication.

A new and a used gudgeon pin; the marks on the used pin are blue in colour and have been caused by overheating because of unduly high bearing pressures

To prevent a larger rise in temperature due to friction, it is essential to restrict the bearing pressures to about 5,500 lb/in^2 on the projected area. However, the allowable pressure can be increased to an absolute maximum of 5,680 lb/in^2, provided there is no doubt that the lubrication is adequate and that the pin deflections are unlikely to be large enough to break through the oil film. If pressures are allowed to exceed this value, the pins may over-heat to such an extent as to go a straw colour or even to blue.

It is usual to radius the inner ends of the holes in the bosses to relieve, in some measure, the stress concentrations that can occur locally both in the boss and in the adjacent parts of the pin. Some manufacturers taper the inner ends of the bores of the cross holes so that the full bearing area is only made use of when the pin is deflected sufficiently to bear in the tapered portion. Others do not consider this to be a good practice, because it reduces the effective bearing area if the pin does not deflect in this way. Moreover, when the bores are tapered there is a tendency for the effective bearing area to be offset further away from the small end, so that heavy bending loads are applied to the pin.

Lubrication of gudgeon pin bearings calls for special attention. It is well known that a rotating journal drags the oil film round with it and, because of the hydrodynamic wedge effect thus obtained, tends to operate under conditions of full fluid lubrication. These conditions do not obtain in gudgeon pin bearings, for the reciprocating nature of the load tends to squeeze the oil film alternately from one side of the pin to the other. In two-stroke engines in which petroil lubrication is employed and in two-stroke diesel units it is advisable to fit bearing bushes in the piston bosses, particularly when cast iron pistons are employed. This is because loading is more continuous in two-stroke than in four-stroke engines. Except in low rated units, petroil lubrication is barely adequate, and in diesel units the loading is too high for unbushed bosses. In diesel engines, it is usual to cut helical grooves in the bushes. These grooves are spaced in such a manner that during the oscillatory cycle of the bearing, lubricant in each wipes the pin to the extent of the full width of the adjacent lands in the bush.

Most two-stroke diesel engine pistons are oil cooled, so it is a relatively simple matter to provide a pressure feed in the piston bosses. In the Ricardo Cocktail Shaker type of piston, vertical passages can be drilled from the bearings to the oil gallery and the inertia loading of the column of oil in these passages provides the pressure feed. In other engines, the lubricant is carried up the connecting rod and into the gudgeon pin whence it is passed out to the bearings in the bosses in the manner already described in the paragraphs dealing with end pads, under the heading "Axial location."

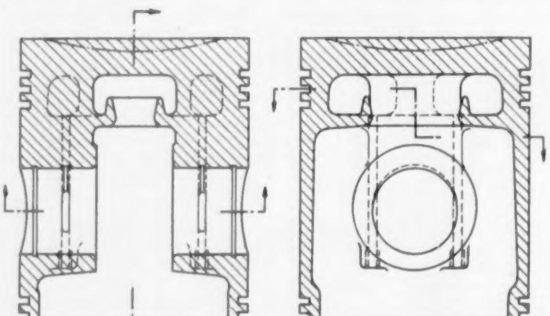
The rate of wear in well maintained two- and four-stroke engines generally is such that the service life before reboring the cross holes and fitting over-size gudgeon pins is necessary, is in the order of 90,000-100,000 miles. This low rate of wear can be obtained only if the oil is kept clean, and it is more likely to be achieved with low expansion alloy pistons because with this material excessive clearances are less likely to develop between the pin and the bore. Accelerated rates of wear can be caused in diesel engines by incorrect fuel pump settings. If the pump timing is too far advanced, steep pressure rises occur in the cylinder, and the resultant hammering on the gudgeon pin can increase the clearance between the pin and the bores of the bosses to as much as $\frac{1}{32}$ in in a relatively short period of time.

Small end bushes are usually of lead-bronze with high tin content, or of phosphor bronze. Some manufacturers now use wrapped steel bushes lined with copper-lead. Where copper-lead is employed, it is essential to line it on steel otherwise it is not strong enough. Running clearances are usually about 0·0003 in, and because this clearance is so small, it is usual to broach and burnish wrapped bushes to size after they have been pressed into the small end. Providing the bearing material is adequately strong, and most are, it should be possible to exceed slightly the figure of 5,680 lb/in^2 quoted for the maximum permissible pressure in gudgeon pin bosses, because the temperature of the small end is unlikely to be as high as that of the bosses.

In two-stroke diesel engines, helical grooving, on the principle already outlined in the paragraph dealing with bushed piston bosses, is frequently incorporated. It is of interest to note that the ends of the bushes, unlike the ends of the bosses, are rarely, if ever, radiused. In view of the fact that the rods may vibrate, or weave, in resonance at certain engine speeds, radiusing of the ends of the bushes might appear to be desirable.

Strength and stiffness

The first stage in the design process is to decide on the overall proportions of the pin and bosses. In general, for normal automobile applications, the pin length should be about 80 per cent of the cylinder bore diameter and the total



In this Ricardo cocktail shaker type, oil-cooled piston, pressure lubrication of the bearing areas between the gudgeon pin and the piston bosses is effected by the inertia loading on the oil in the vertical passages drilled in each side of the struts and bosses

bearing length of the two bosses should be about 33⅓ per cent of the bore. Given these dimensions, it is then necessary to calculate the pin diameter that will restrict the bearing pressures on the projected area to the figure of 5,500-5,680 lb/in^2, already mentioned. For all strength calculations, the load is assumed to be the estimated maximum combustion pressure multiplied by the plan area of the piston. This gives only a slightly conservative value because, although inertia loads relieve those due to gas pressure, slow speed running under full load conditions has to be considered.

It is generally thought to be necessary to limit deformation of the pin to 0·001 in to avoid breaking through the oil film. Providing this requirement is fulfilled, the pin is usually strong enough. However, it is desirable also to check the shear and bending stresses by the well known formulæ, except where experience has shown that the sections are obviously adequate for the duty they have to perform. In all strength calculations it is safest to work on the core diameter, that is, to ignore the case because of the possibility of there being hardening cracks in it. The pin is subjected simultaneously to three types of loading and, of course, their combined effects have to be considered. The three types of stresses are: shear on the planes between the small end and the inner faces of the bosses, bending due to the connecting rod thrust and its reactions on the bosses, and bending due to the tendency of the pin under load to deform from the truly circular to an oval section.

A rough finish in the bore of a gudgeon pin often causes fatigue failure of this type

The formula for obtaining bending deflections is well known and need not be quoted here, but the lesser known one, for calculating deformation from the circular to oval section is:

$$\text{Deformation } w = 0{\cdot}416\frac{P_m D^3}{LEh^3}$$

where P_m = total load (lb)
D = outside diameter of pin (in)
L = length of pin (in)
E = modulus of rigidity, 30×10^6 (lb/in^2)
h = difference between inside and outside diameters (in).

When bushes are fitted, it is also necessary to check the bursting stress due to their interference fit in the gudgeon pin bosses. This stress should be limited to about ⅛th of the ultimate tensile strength of the material of the bosses, so that an ample margin of strength is left to cater for additional loads due to gas pressure and inertia forces. The formulæ for checking bursting stresses are derived from Lamé's theory for thick cylinders, and are as follows:

$$P = \frac{\delta}{\frac{b}{E_b}\left[\frac{1+\left(\frac{b}{c}\right)^2}{1-\left(\frac{b}{c}\right)^2}+U_b\right]+\frac{b}{E_p}\left[\frac{1+\left(\frac{a}{b}\right)^2}{1-\left(\frac{a}{b}\right)^2}-U_p\right]}$$

$$\text{and } f = P\left[\frac{c^2+b^2}{c^2-b^2}\right]$$

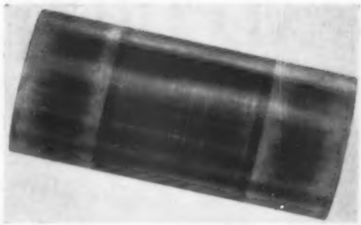
where a = inside radius of pin
b = outside radius of pin
c = outside radius of boss
δ = interference on the radius
P = contact pressure
E_b = modulus of elasticity of the boss (assumed 10×10^6 lb/in^2 for Y-alloy)
E_p = modulus of elasticity of the pin (assumed 30×10^6 lb/in^2)
U_b = Poisson's ratio for the boss
U_p = Poisson's ratio for the pin
f = Hoop or bursting stress in the boss

For low expansion aluminium alloy, $U=0{\cdot}25$ and $E=10{\cdot}03\times10^6$.

For steels, $U=0{\cdot}25$ and $E=30\times10^6$.

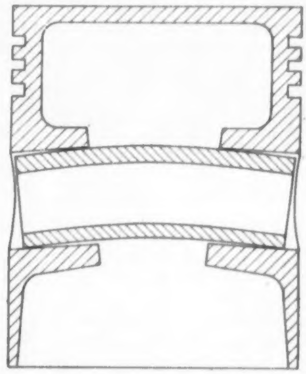
For lead-bronze, $U=0{\cdot}34$ and $E=12\times10^6$.

For Y-alloy, $U=0{\cdot}25$ and $E=10\times10^6$.



The bright markings round the periphery of this 2 in diameter gudgeon pin are thought to be an indication that bending deflection has occurred under load. The darker markings are of straw colour and have been caused by overheating

A factor that has considerable influence on the strength of a pin is, of course, its bore shape and, as has already been mentioned, a poor surface finish can lead to fatigue failure. Until about 12-15 years ago, bores of pins were generally tapered from the ends, either to the centre or to a point just outboard of the inner face of each boss, the thickest section being in the centre. The second arrangement is considered to be the better since it provides a thick section on the planes which are most highly stressed, that is, between the ends of the small end and the inner faces of the piston bosses. However, the practice of tapering the bores has been abandoned except in high speed, sports and racing engines where light weight is of prime importance. So far, no reliable method has been evolved for calculating the deformation from the truly circular to the oval shape for pins of this type. When the bores are tapered, it is safest



Failures can often be attributed to bending deflection of the gudgeon pin, which is exaggerated in this diagram. From this it can be seen how the markings shown in the half-tone illustration at the top of this column were effected

to calculate the deformation for a parallel bore pin, the wall thickness of which is equal to the minimum section of the tapered bore. For normal applications, parallel bore pins are generally regarded as most satisfactory from both the design and production points of view.

Pin failures usually take the form of a fracture running the length of the pin. If the design is sound, these fractures are generally caused by a fault in the material or by fatigue due to stress concentrations set up by unsuitable bore finish. Failures are sometimes experienced in the planes between the small end and the bosses. Again, these usually originate at notches due to tool marks or other defects in the bore. A typical failure is shown in the accompanying illustration. A crack originated somewhere near the centre of the pin and spread longitudinally. When it reached the plane between the small end and the bosses it spread circumferentially under the influence of the principal stresses in this area.

Acknowledgment is made to T. O. Hunt, Technical Director of Specialloid Ltd., for information and illustrations used in this article.

ALCOHOL IN DIESEL ENGINES*

The Utilization of Power Alcohol in Combination With Normal and Heavy Fuels in High Speed Diesel Engines

By Dr. H. A. Havemann, M. R. K. Rao, A. Natarajan and T. L. Narasimhan

THE development of a method of using Power Alcohol in combination with cheaper, low grade fuels is interesting from a technical point of view, since it makes practicable the employment of heavier fuels in high speed diesel engines. It is also interesting because, in some countries, Power Alcohol is available in considerable quantities as an indigenous fuel, for which the opening of an additional market may be a progressive step economically.

This investigation was therefore carried out to determine the advantages and disadvantages of using alcohol in some engines of common design having a high compression ratio and accepting alcohol in proportions much greater than are employed at present. It was also aimed at overcoming the limitations set by the poor water tolerance of blends of alcohol with most hydrocarbon fuels. Another problem that was receiving serious consideration at the same time, and originally independently, was the utilization of heavy hydrocarbon fuels in high speed diesel engines. Heavy hydrocarbon fuels are generally employed in slow speed engines and are cheaper than the oils for high speed units.

However, heavy fuels have two main disadvantages, and these make them unsuitable for use in high speed diesel engines: they are the slowness of the combustion process, which results in a smoky exhaust, and the high sulphur content, which leads to excessive corrosive wear. It was therefore decided that if a method were discovered that made possible the use of these fuels in high speed diesel engines, it would constitute a step forward economically and scientifically.

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Review of previous work

Much research has been conducted by a number of workers to find ways and means of using the heavier and residual fuels in place of superior distillates. These activities can be broadly classified under two headings:

1. Improving the ignition quality of the fuel.
2. Minimizing the deleterious effects of high sulphur and carbon residues.

Ignition quality of the fuel: Regarding the ignition quality of the fuel, Ricardo[1] has shown that it could be improved by the addition of small proportions of certain chemical substances. These additives are supposed to expedite the ignition of the fuel by reducing the temperature of self-ignition. Amyl nitrate, ethyl nitrate and acetone peroxide are a few of the many ignition accelerators that are found suitable for blending with diesel fuels of low octane value. One of the practical objections to these accelerators is their high cost. Another is their effect on engine life, but little is known about this aspect. As the proportion of additive is increased, the accelerating effect of each increment decreases.

A second line of attack on the problem of improving combustion is by supercharging the engine. In a recent address, Ricardo[2] has given particulars of tests that he had carried out on an engine running at 1,500-2,000 r.p.m. With a pressure boost of half an atmosphere, the engine ran smoothly on a fuel having a cetane number as low as 18. This may be a good remedy but it will also increase the cost of the basic engine unit as well as the cost of maintenance. The air utilization factor, however, apparently does not vary a great deal.

McLaughlin[3] and his associates report that combustion in a high speed diesel engine could be accelerated by carburetting volatile fuel of a high self-ignition temperature. Advantage can be taken of this phenomenon either to reduce the exhaust smoke density or, for the same smoke density, to boost the engine power by some 20 per cent. Maxwell, of the Caterpillar Corporation, found that by carburetting mixtures of alcohol and water in a supercharged diesel engine, the efficiency could be slightly improved, as also could the power output.

Young, of the Sinclair Research Laboratories, found that fuels requiring a compression ratio of 25:1 for correct combustion could be made to burn satisfactorily at a compression ratio of 12:1 by carburetting some 5 per cent of *n*-Heptane. By a similar method, the authors have found it possible to use distillate fuels with a cetane value as low as 12, and the performance is as good as that of diesel fuels having a cetane value of 50. H. M. Gadebusch, of General Motors, has also reported similar results. Hobbs[4] has found that, when added to a normal diesel fuel, alcohol tends to reduce shock loading, on ignition, improves combustion and reduces smoke density. Aubert, when using diesel oil and ethanol in a dual injection engine, obtained similar results.

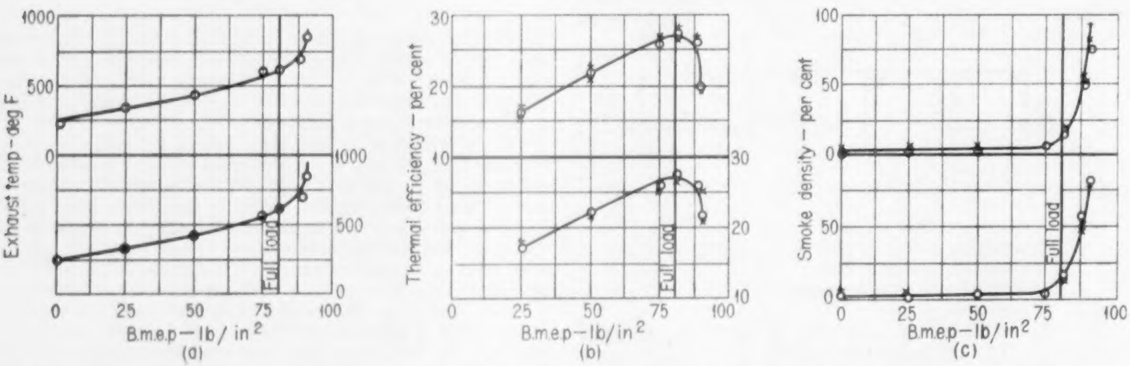

Fig. 1. Ricardo E6 engine: compression ratio: 21·22: 1. Speed 1,250 r.p.m. Fuels: Grade A and Grade B oils, and blends of these oils with alcohol. Injection timing: 37 deg B.T.D.C. The lower curves in a, b and c were obtained with Grade A oil and blends, and the upper ones with Grade B oil and blends

+ neat fuel × with 5 per cent blend o with 10 per cent blend

Utilization of power alcohol in high speed diesel engines

Deleterious effects of residual matter: With regard to the deleterious effects of high sulphur and carbon contents in heavy fuels, Brewer and Thorp[5] have shown that, for medium speed engines, the use of heavier fuels leads to injector nozzle encrustation, and cylinder wear and crankcase fouling are slightly increased. J. P. R. Smith[6] has also noticed similar effects when medium speed engines are run on heavy fuels. In large, slow speed engines, Lamb[7] reports that there is little or no increase in the rate of liner wear or carbon deposit resulting from a change to residual fuel. On the other hand, in a high speed engine developing 5 b.h.p. at 1,500 r.p.m., A. Natarajan and M. R. K. Rao[8] have found that there is a noticeable increase in the wear of the liner and top ring, as well as an increase in the carbon deposit when the fuel is changed from B.S.S. Grade A to B.S.S. Grade B fuel. The tendency for ring sticking is also more pronounced.

Broeze and Wilson[9] report that sulphur in the fuel aggravates the liner wear problem. C. H. Cloud and A. J. Blackwood,[10] of Esso Laboratories, have obtained the same results. It is suggested that the sulphur in diesel fuel burns to sulphur trioxide, which raises the dew point and causes condensation of aqueous acid products at a higher temperature than is normally expected. This leads to increased liner wear and sludge formation. A reduction in the rate of wear can be obtained by chromium plating liners or piston rings and by using heavy duty additive type lubricants.

Conclusions from previous work: The foregoing review can be summarized thus:

1. It has been found in practice that heavier fuels, provided they are preheated and purified, can be burned successfully in large and medium size engines without the addition of ignition accelerators. However, in the case of high speed engines it is necessary that some sort of combustion accelerator should be employed in addition to preheating and purifying the fuel.
2. Because of economic and other considerations, chemical accelerators have not yet become a practical success.
3. Carburation of suitable quantities of volatile fuels of high self-ignition temperature leads to combustion acceleration and power boosting. This has the additional advantage of a higher air utilization factor.
4. To some extent, blending gasoline or alcohol with diesel fuel leads to improvements in combustion and engine performance.
5. Heavier fuels lead to higher rates of carbon deposit and wear, but combustion accelerators may reduce these defects.
6. Chromium plating liners or top rings and the use of heavy duty lubricants reduce cylinder wear.

Scope of the work presented

The foregoing review shows that heavier hydrocarbon fuels can be used in high speed diesel engines only if a suitable ignition or combustion accelerator is used. The works of Hobbs, Aubert, McLaughlin and others suggest that alcohol could be used successfully as a combustion accelerator. While Hobbs used a blend of alcohol and diesel fuel in an ordinary engine, Aubert applied a dual injection system and the others used carburettors, etc., for introducing volatile fuels mostly for the purpose of boosting the power output. Consequently, only a low percentage of volatile fuel was introduced. The aim of the investigation described here was, however, to find a way of using alcohol as the principal fuel.

TABLE 1. RESULTS OF EXPERIMENTS REGARDING MISCIBILITY OF DIESEL FUELS WITH ALCOHOL

(With furnace oil and alcohol there was no miscibility either at room temperature or elevated temperatures)

Oil cm^3	Alcohol cm^3	Room Temperature	Heated to 60 deg C	Few Drops Water Added
Grade A				
20	5	4·5 cm^3 alcohol separated	Clear solution	Separation takes place
20	10	11 cm^3 Grade A oil separated	,,	,,
20	15	16 cm^3 Grade A oil separated	,,	,,
20	20	14 cm^3 Grade A oil separated	,,	,,
5	20	1 cm^3 Grade A oil separated	,,	,,
10	20	4 cm^3 Grade A oil separated	,,	,,
15	20	14 cm^3 Grade A oil separated	,,	,,
Grade B				
20	5	4·5 cm^3 alcohol separated	1·5 cm^3 alcohol separated	Separation takes place
20	10	8 cm^3 alcohol separated	Clear solution	,,
20	15	12 cm^3 alcohol separated	7 cm^3 Grade B oil separated	,,
20	20	13 cm^3 alcohol separated	9 cm^3 Grade B oil separated	,,
5	20	Turbid solution	Turbid solution	
10	20	,,	,,	
15	20	,,	,,	

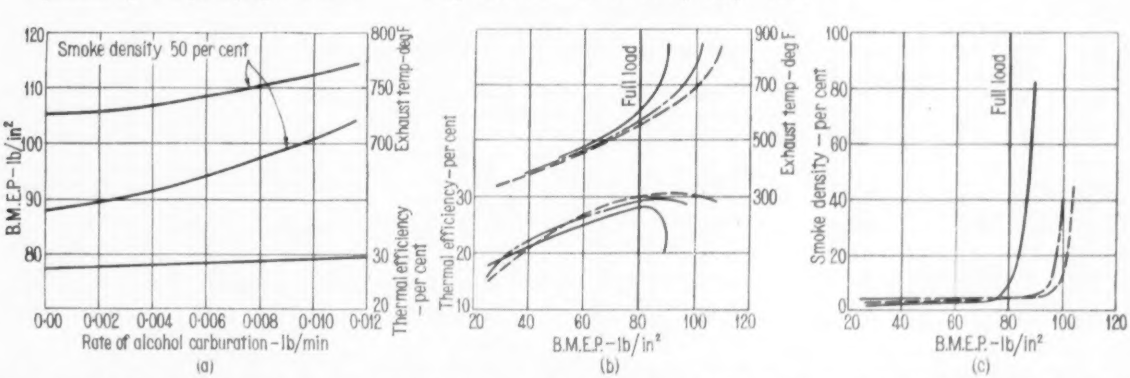

Fig. 2. Ricardo E6 engine run under the same conditions as in Fig. 1 except that the injected fuel is Grade B oil with and without alcohol carburation

——————— Grade B oil. — — — — — — — — Grade B oil with 0·01 lb/min alcohol carburation

- - - - - - - - - Grade B oil with 0·012 lb/min alcohol carburation

TABLE 2. SPECIFICATIONS OF THE RICARDO E-6/R VARIABLE COMPRESSION ENGINE. SERIAL No. 21/46

General details	Four-stroke, compression ignition, vertical, water cooled
Number of cylinders	1
Bore diameter	3 in
Stroke	4½ in
Swept volume	506 cm^3
Compression ratio	21·22 : 1
Rated power	4·2 b.h.p. at 1,250 r.p.m
Tappet clearance: inlet	0·006 in
exhaust	0·008 in.
Inlet valve opens	9 deg B.T.D.C.
Inlet valve closes	37 deg A.B.D.C.
Exhaust valve opens	44 deg B.B.D.C.
Exhaust valve closes	5 deg A.T.D.C.
Fuel injection pressure	100 atm
Fuel injection timing	37 deg B.T.D.C.
Injector	C.A.V. type DN 12 SD 12
Fuel pump	C.A.V. type PEIB 5·5 mm diameter plunger
Combustion chamber	Ricardo Comet Mk II—compression swirl

In addition to acting as a combustion accelerator, alcohol was expected to act as a thinning agent that would reduce the viscosity of heavy fuels to a value suitable for the injection equipment. The Hobbs method of blending alcohol with heavier fuels was tried out first. However, alcohol-diesel blending did not produce the expected results, so it was decided to change over to carburation of alcohol and injection of diesel fuel. This stage of the work was more successful.

Alcohol as a blend with diesel fuel in high speed diesel engines

Preparation of blends of alcohol-diesel fuels: It is common experience that (1) anhydrous alcohol can combine in any proportion with hydrocarbon fuels, (2) such solutions have a very low water tolerance, (3) hydrocarbons of the paraffin series are much less readily soluble than those of the aromatic series and (4) the solubility decreases with increase in molecular weight. These conditions hold good for non-volatile hydrocarbon fuels as also for volatile hydrocarbon fuels. If anything, the difficulties of blending are increased in the case of diesel fuels because the molecular weight gradually increases with weight of fuel, from distillate gas oil to residual oil. In the Table 1 details are given of the experimental results obtained regarding miscibility of B.S.S. Grade A, B.S.S. Grade B and furnace oils.

These experiments indicate that Grade A oil is miscible with alcohol to a maximum extent and mixing is facilitated by heating the blend to about 60 deg C. However, traces of water added to the blend, whether cold or hot, separate the components. With the Grade B oil blend, miscibility is lower and presence of water is more harmful. In the case of furnace oil there is little or no miscibility. Within these limitations of miscibility blends were prepared containing 10 per cent and 5 per cent alcohol and the rest Grade A or Grade B oil.

Engine tests with alcohol-diesel fuel blends: These blends were used as fuels first in a Ricardo Research Engine with a diesel head and later in a Kirloskar-Petter, high speed diesel engine with a precombustion chamber. The results of the Ricardo engine trials are presented graphically in Figs. 1a, 1b and 1c. The tests revealed that there was no noticeable improvement in combustion, exhaust smoke density or in the running characteristics of the engine. The results from the Kirloskar-Petter engine trials are not shown, since they were similar to the others.

Alcohol as a supplementary fuel in high speed diesel engines

The results obtained with alcohol-diesel fuel blends, although good enough in certain respects, were not exactly what was expected. It was therefore decided to try carburation of alcohol with injection of different diesel fuels.

Tests with Ricardo Research Engine: These tests were limited to injection of Grade B diesel fuel and carburation of alcohol. Table 2 gives the specifications of the Ricardo Research Engine. The unit was fitted with the precombustion chamber diesel head, which was locked so that the apparent compression ratio was 21·22 : 1. Fuel was injected through a pump incorporating a device to adjust the injection timing, and alcohol was introduced through a variable jet carburettor into the induction manifold. Each fuel system incorporated a separate flowmeter to measure consumption, and the air flow was measured by Alcock viscous flow, air meter. A C.R.C. photoelectric smoke meter was used to measure the smoke density. Power measurements were made with a Farnborough electric engine indicator and an electric dynamometer. Other standard instruments were used to measure the speeds, temperatures, etc.

Test procedure: The engine was started on diesel fuel and allowed to warm up thoroughly. It was then loaded to a predetermined value. The injection timing was set at 37 deg B.T.D.C. for optimum results. Cooling water temperature was maintained at 70 deg C at the outlet from the engine. At steady running conditions one set of the following readings was taken:

Air and fuel flow, speed, brake load, indicator diagram, temperatures, smoke meter and barometer.

Maintaining the load constant, the supply of diesel fuel was gradually reduced, and alcohol inducted in suitable proportions. For each combination of alcohol and diesel fuel, the engine was run at steady conditions and the various observations were recorded as before. Incipient knocking indicated the upper limit to which the propor-

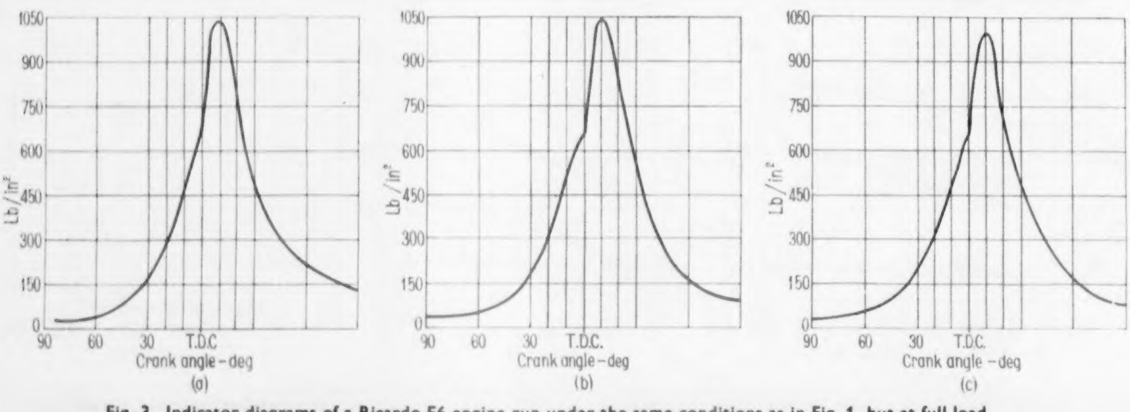

Fig. 3. Indicator diagrams of a Ricardo E6 engine run under the same conditions as in Fig. 1, but at full load
a With Grade B oil fuel injected. b With Grade B oil fuel injected and 0·01 lb/min alcohol carburetted
c With Grade B oil fuel injected and 0·012 lb/min alcohol carburetted

tion of alcohol could be increased. This procedure was repeated for different loads.

Test results: Test results are graphically presented in Figs. 2a, 2b, 2c, and 3a, 3b and 3c, and summarized as follows:

1. The maximum quantity of alcohol that could be inducted was 36 per cent of the fuel required by the engine at full load. This was found to be equivalent to an air : alcohol ratio of 54 : 1. It was also noticed that the engine began to misfire at idling if the mixture was made richer than 54 : 1.
2. As a result of induction of alcohol, the engine could be overloaded by more than 16 per cent, and under overload conditions, the smoke density was no more than that at full load with neat diesel fuel, Figs. 2a and 2b.
3. A direct inference from these overload characteristics was that the air utilization factor was higher with alcohol induction, as compared to normal running conditions.
4. Except at very light loads, there was a small but distinct increase in thermal efficiency as a result of the induction of alcohol. This value increased as the percentage of inducted alcohol was increased, Figs. 2a and 2b.
5. There was no perceptible change in the volumetric efficiency.
6. The smoke density was reduced considerably as a result of the induction of alcohol. For any particular load the higher the rate of alcohol inducted, the lower the smoke density, Figs. 2a and 2c. At full load, with induction of maximum quantity of alcohol, the exhaust colour was comparable with that of a petrol engine.
7. There was a distinct drop in the exhaust gas temperature when alcohol was inducted, Figs. 2a and 2b.

Tests with a Petter engine: Because of the promising results obtained from the Ricardo engine trial, the investigation was extended to a production engine with an open combustion chamber. For this purpose, a Petter AV1 series II engine, developing 5 b.h.p. at 1,500 r.p.m., was chosen. Detailed specifications of the engine are given in Table 3.

Fuel control was effected by a centrifugal governor. A three-way cock was fitted to the fuel inlet of the injection pump so that it was possible to change over easily from one type of fuel to another. Heavy fuel was preheated in a water bath prior to its admission to the injection pump. This bath was located as close to the pump as practicable so that there was no chance for the fuel to cool after leaving it. A variable jet, industrial carburettor, through which the alcohol was inducted, was fitted to the inlet manifold.

Flow-meters were used to measure the consumptions of the fuels, and air flow was measured first with an air box, and later checked with an Alcock flowmeter. Exhaust gas temperature was measured as close to the engine as possible, and a C.R.C. photoelectric smoke-meter was used to estimate the smoke density. A rough estimate of the extent of the carbon particles in the exhaust gas was made by collecting them on a glass plate which was then photographed. The cooling water temperature was maintained at about 80 deg C throughout the tests. Power measurement was effected by means of a water brake directly coupled to the engine.

The fuels chosen for the tests were:

1. B.S.S. Grade A diesel fuel, fulfilling the makers' specifications for use with the engine
2. B.S.S. Grade B diesel fuel, an inferior grade suitable for slow speed engines
3. Furnace oil, used mostly in boiler practice
4. Power Alcohol, produced by Mysore Sugar Factory, Mandla.

Detailed specifications of these fuels are given in Table 4.

Fuels B.S.S. Grade A and B flowed freely at room temperature; they were supplied to the injection pump after filtration. Furnace oil was too viscous at room temperature for use with the injection equipment. It was therefore preheated to 90 deg C and passed through three felt filters before it was transferred to the engine fuel tank. This fuel was again heated in a water bath to about 95 deg C before it was passed to the injection pump. The specific gravity of this fuel at various temperatures was also determined experimentally and was used in the calculation of fuel consumption.

Test procedure: The engine was started and allowed to warm up on diesel fuel for half an hour. It was then loaded to a predetermined value. With the engine running steadily, the following readings were taken: (1) Speed of the engine, (2) Brake load, (3) Fuel consumption, (4) Air consumption, (5) Ambient air temperatures and pressure and humidity, (6) Exhaust gas temperature, (7) Smoke density, (8) Carbon particles in the exhaust gas were collected on a glass plate for 15 seconds and then photographed.

With the brake load constant, alcohol was admitted by stages into the engine, while the fuel pump governor automatically reduced proportionately the fuel oil injected. As the percentage of alcohol inducted increased, a stage was reached when the engine started missing and hunting. This was the

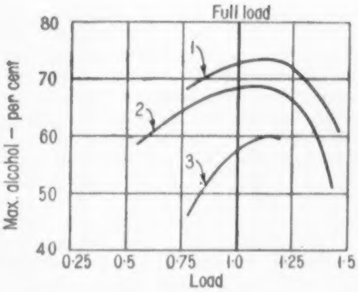

Fig. 4a. Petter engine: Compression ratio 16·5 : 1. Speed 1,500 r.p.m. Injection timing 27 deg B.T.D.C.
1 Grade A oil. 2 Grade B oil
3 Furnace oil

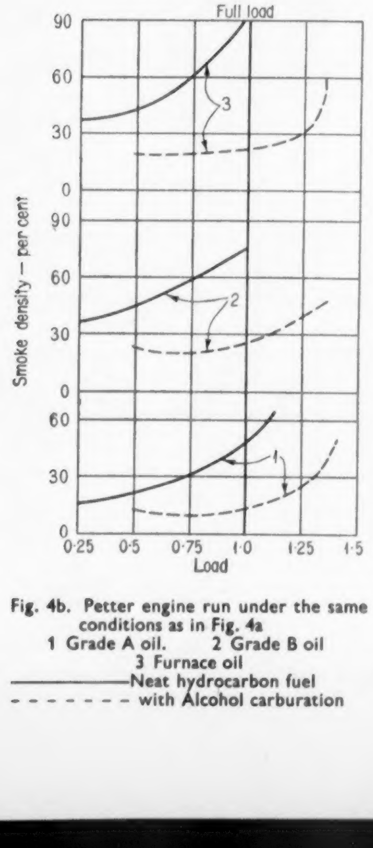

Fig. 4b. Petter engine run under the same conditions as in Fig. 4a
1 Grade A oil. 2 Grade B oil
3 Furnace oil
———— Neat hydrocarbon fuel
- - - - - - - with Alcohol carburation

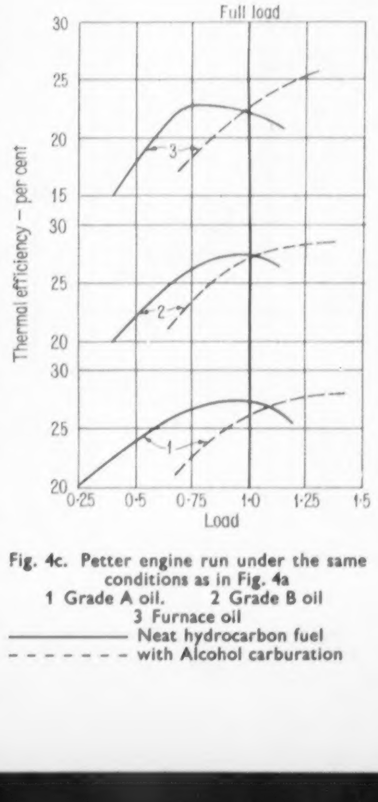

Fig. 4c. Petter engine run under the same conditions as in Fig. 4a
1 Grade A oil. 2 Grade B oil
3 Furnace oil
———— Neat hydrocarbon fuel
- - - - - - - with Alcohol carburation

maximum limit for the admission of alcohol. For each combination of alcohol and diesel fuel proportions, one set of observations was recorded as before, the only additional observation being the alcohol flowmeter reading. This procedure was repeated for half, three-quarters, full and overload conditions. The "full load neat fuel smoke density" was the limit up to which the engine was overloaded for each fuel.

Test results: The test results are graphically presented in Figs 4a, 4b, 4c, 5, 6 and 7, and summarized below:

1. Under full load conditions nearly 70 per cent of alcohol could be inducted in combination with Grade A fuel. On the other hand, under the same load conditions, only 60 per cent of alcohol could be inducted in combination with either Grade B fuel or furnace oil. Under overload conditions slightly lower percentages of alcohol could be inducted, Fig. 4a.
2. With alcohol induction, the engine could be overloaded to about 45 per cent and 40 per cent respectively with Grade A and Grade B fuels. Under overload conditions the smoke density was no more than that at full load with neat diesel oil. When running on neat furnace oil, however, it was found that the engine could be loaded only to 50 per cent for passable smoke density, Fig. 4b.
3. A direct inference from result (2) was that the air utilization factor was higher with alcohol induction, as compared with normal running conditions.
4. Under full and part load conditions, there was generally a drop in the thermal efficiency of the engine when run on any of the diesel fuels in combination with alcohol carburation. On overload, there was a definite improvement in thermal efficiency, Figs. 4c and 5.
5. There was a slight improvement in the volumetric efficiency.
6. The smoke density was reduced considerably as a result of induction of alcohol. For any particular load, the higher the rate of alcohol inducted, the lower was the smoke density, Fig. 6. When the engine was run on Grade A and Grade B fuels at full load with induction of maximum quantity of alcohol, the exhaust colour was comparable with that of a petrol engine.
7. Exhaust temperature generally increased slightly with the induction of alcohol except when furnace oil was used; with this oil it decreased slightly at higher loads, Fig. 7.
8. The proportion of free carbon particles in the exhaust gas was lower with alcohol induction. This was very pronounced in the case of furnace oil.

Alcohol - water solution as supplementary fuel: A second set of experiments was conducted subsequently to study the influence of water dissolved in alcohol on the performance of the engine. For this purpose, solutions of water and alcohol were prepared in different proportions and tried in combination with Grade B oil. The observations are given in Table 5. The results can be summarized as follows:

1. Up to 30 per cent of water could be mixed with alcohol without any deterioration in performance of the engine.
2. There was not much variation in thermal efficiency.
3. There was little difference in the

TABLE 3. SPECIFICATIONS OF PETTER AV1 SERIES II ENGINE

General details	Four-stroke, compression ignition, vertical, cold starting, water cooled
Number of cylinders	1
Bore diameter	80 mm (3·15 in)
Stroke	110 mm (4·33 in)
Swept volume	553 cm³ (33·73 in³)
Compression ratio	16·5 : 1
Rated power	5 b.h.p. at 1,500 r.p.m.
Tappet clearance	0·007 in
Inlet valve opens	4½ deg B.T.D.C.
Inlet valve closes	35½ deg A.B.D.C.
Exhaust valve opens	35½ deg B.B.D.C.
Exhaust valve closes	4½ deg A.T.D.C.
Fuel injection pressure	2,500 lb/in²
Fuel injection timing	24 deg B.T.D.C.
Nozzle	Three-hole, 0·24 mm diameter by 1·75 mm long stem, No. HL-S24 C 175 P 3
Fuel pump	Bryce, type A1 AA 70/55.99 No. AA 32967
Combustion chamber	Open
Fuel oil	A high grade light distillate diesel fuel in accordance with B.S.S. No. 209/1947, Class A

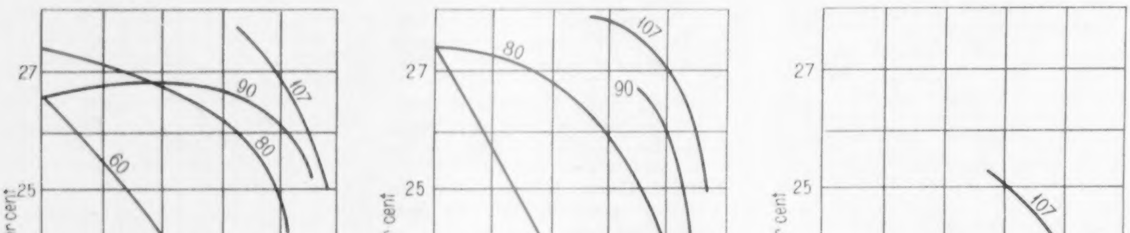

Fig. 5. Petter engine: Compression ratio 16·5: 1. Speed 1,500 r.p.m. Injection timing 27 deg B.T.D.C. Full load (80 lb/in² b.m.e.p.)
a Grade A oil b Grade B oil c Furnace oil

exhaust temperature. However, the trend was generally towards lower values as compared with pure alcohol.

4. There was no perceptible variation in smoke density

Discussion of results

The Ricardo engine is much less sensitive to the grade of fuel supplied, while the Petter engine is intended to run on those fuels that come within B.S.S. 209/1947 Grade A. With the Ricardo engine, the maximum amount of alcohol that can be carburetted into the engine is limited by the occurrence of knocking, whereas in the case of the Petter engine, it is limited by missing and hunting. Under all conditions of running with alcohol, the Petter engine was free from knock.

It has been noticed that, whatever the load condition, as long as the air:alcohol ratio is greater than 54:1 there will be no knocking in the Ricardo engine. As soon as the mixture is made richer, knocking starts. This may be due to the fact that enrichment of the mixture reduces the spontaneous ignition temperature and leads to auto-ignition, particularly where the compression ratio and hence the compression temperature are high. On the other hand, with the Petter engine, the compression ratio and compression temperature are of a lower value. Because of this fact, however much the mixture is enriched, there is no possibility of self-ignition and consequent knocking. This is further substantiated by the fact that the Petter engine began to knock violently when 73 octane petrol, having a lower self-ignition temperature than alcohol, was inducted into the engine. From these considerations, it may be concluded that the main condition to be satisfied by an inducted fuel is that it must have a high self-ignition temperature.

It should also be noted that in the Petter engine, the enrichment of the mixture beyond the maximum limit causes missing and hunting under all load conditions. The cause for this is not clear and is a matter for further investigation. From the curves in Fig. 8 it can be seen that for any particular air:alcohol ratio the quantity of diesel fuel injected per cycle will be a minimum with Grade A fuel and a maximum with furnace oil. This indirectly indicates that with Grade A fuel, a maximum amount of alcohol can be utilized in the engine, and this quantity is reduced as the fuel becomes heavier.

The diesel engine, when run on carburetted alcohol with any of the three hydrocarbon fuels, gives a higher air utilization factor. Consequently, it can be overloaded to nearly 40-45 per cent when Grade A oil is used and 35-40 per cent when Grade B oil is employed. The greater the amount of alcohol carburetted, the higher is the overload that the engine can take.

In the Ricardo engine, running on Grade B oil with alcohol induction, there is a certain, though small, increase in the thermal efficiency of the engine. Figures 3a, 3b and 3c show the indicator diagrams taken at full load, on the Ricardo engine with Grade B oil alone and with Grade B oil and alcohol inducted at the rates of 0·01 lb/min and 0·012 lb/min respectively. The greater the amount of alcohol carburetted, the steeper the pressure-crank angle curve; combustion tends to be completed earlier and the peak pressure is reached earlier in the cycle: hence the increase in thermal efficiency of the engine. This may

TABLE 4. PROPERTIES OF DIESEL OILS, FURNACE OIL AND ALCOHOL USED IN THE EXPERIMENTS

	Grade A oil	Grade B oil	Furnace oil	Alcohol
Sp. gr. at 75 deg C	0·84	0·87	0·90	0·78
Cetane number	45	40	—	—
Viscosity at 100 deg F, Redwood I secs	35	45	390	—
Carbon residue Conradson, per cent weight	0·05	1·0	4·8	—
Sulphur, per cent weight	0·3	1·2	2·5	—
Water, per cent weight	Zero	0·05	0·1	Less than 0·5
Sediment, per cent weight	Zero	0·01	0·01	—
Ash, per cent weight	Zero	0·01	0·01	—
Calorific value (lower) B.Th.U/lb	18,600	18,100	18,150	11,600

Fig. 6. Petter engine run under the same conditions as in Fig. 5
a Grade A oil b Grade B oil c Furnace oil

TABLE 5. OBSERVATIONS OF TESTS WITH ALCOHOL-WATER SOLUTION AS A SUPPLEMENTARY FUEL IN THE PETTER ENGINE

Load: Full load — Injected fuel: Grade B oil

Per cent water in alcohol	Exhaust Temperature	Smoke density per cent	Thermal efficiency	Per cent alcohol
0	720 deg F	10	25	64
5	720 deg F	10	23·9	60
10	680 deg F	10	25	62
15	700 deg F	15	25·5	60·5
20	710 deg F	28	25·6	58·5
25	680 deg F	30	26·2	54·5
30	690 deg F	15	25·5	55

also account for the incidence of knocking with higher rates of alcohol induction than 0·012 lb/min.

With the Petter engine, the thermal efficiency generally decreases slightly with the induction of alcohol. But, at overloads a higher thermal efficiency is recorded. The exact cause for the drop in the thermal efficiency at part loads is not clear.

There is no appreciable difference in the volumetric efficiency of the engine due to the induction of alcohol. From Figs. 2a, 2c and 6 it can be seen that for constant load, the smoke density is always reduced with the induction of alcohol. This appears to be due to (1) a smaller quantity of hydrocarbon fuel burned per cycle, (2) all the fuel being injected during the early part of the cycle and (3) the rapid combustion of alcohol leading to higher turbulence, all of which are conducive to better combustion of the hydrocarbon fuels. The quantity of hydrocarbon fuel burned per cycle in the engine increases at overload, and the smoke density also increases proportionately.

The presence of free carbon particles in the exhaust is reduced considerably with the induction of alcohol. When the combustion of injected fuel is predominantly one of oxidation of products of destructive decomposition, there are greater chances of the fuel cracking and forming carbon particles. On the other hand, the combustion of alcohol is predominantly a process of hydroxilation and the chances of the fuel cracking are negligible. Consequently, induction of alcohol reduces the quantity of carbon particles in the exhaust gases. The smoke densities obtained when the Petter engine was run on furnace oil alone, and on furnace oil and alcohol, were 90 per cent and 30 per cent respectively.

As mentioned elsewhere, the principal disadvantage of alcohol-diesel fuel blends is their low water tolerance. Carburation of alcohol successfully overcomes this disadvantage. With this method of introducing alcohol, as much as 30 per cent of water in solution with alcohol does not seem to have any

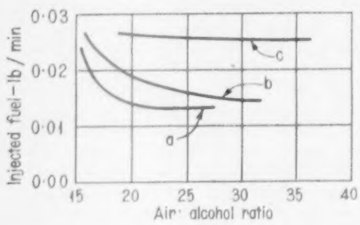

Fig. 8. Petter engine. Compression ratio 16·5 : 1. Speed 1,500 r.p.m. Injection timing 27 deg B.T.D.C.
a Grade A oil. b Grade B oil
c Furnace oil

adverse effect on the performance of the engine. The common standard of 95 per cent alcohol, which can be produced easily, is suitable for the purpose of induction into high speed diesel engines.

Conclusions

1. Alcohol can be used as a power booster fuel in a high compression precombustion chamber engine. It can be used as a primary fuel in open combustion chamber high speed diesel engines, both for normal running and high power boost.
2. Under overload conditions, better air utilization is obtained.
3. For periodic boosting of transport or marine high speed diesel engines this promises to be a better, cheaper and simpler alternative to supercharging.
4. Induction of alcohol invariably results in a cleaner exhaust, especially at higher power output.
5. The exhaust temperature when alcohol is introduced is higher in the case of Grade A and Grade B oils and lower with furnace oil.
6. The presence of appreciable proportions of water in the alcohol does not affect the performance of the engine, so it is not necessary to specify anhydrous alcohol.
7. Further research will be needed to complete the investigation. Details of the work still to be carried out are given in the paper as published in the *Journal of the Indian Institute of Science*.

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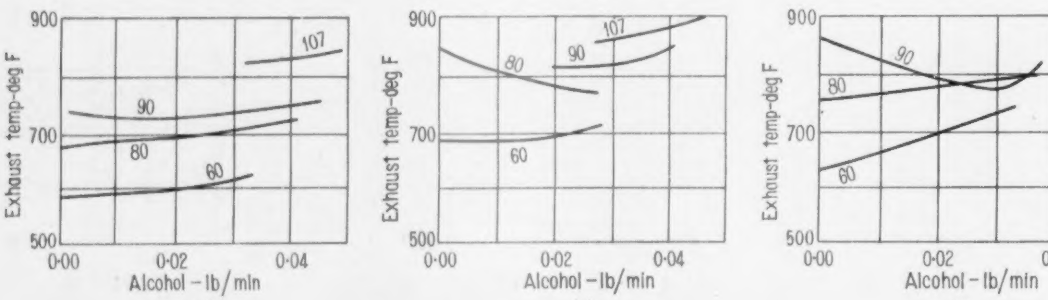


Fig. 7. Petter engine run under the same conditions as in Fig. 5
a Grade A oil b Grade B oil c Furnace oil

TOOL COOLING BY CO_2

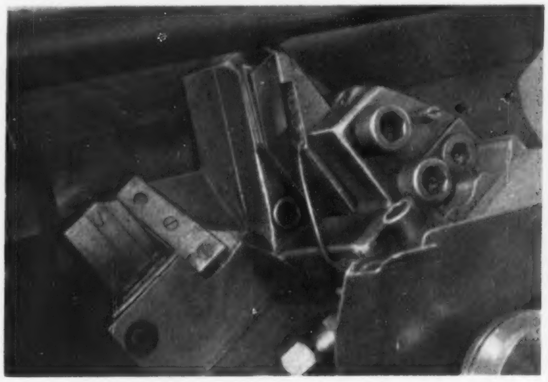
Development of the CeDeCut Technique

GENERAL principles governing the use of carbon dioxide (CO_2) as a tool coolant in various machining operations were outlined in an article in a recent issue.* Necessarily, after the introduction of a new method or process, some time must elapse before a detailed technique can be established for general practical application. Sufficient operating experience has, however, been accumulated in what may be regarded as the development period to justify a report of progress. At present the companies responsible, the Carbon Dioxide Co. Ltd. collaborating with the Central Research Department of the Distillers Co. Ltd., have set up a well-equipped experimental workshop and some two dozen firms are using the process on specific jobs in their own shops. The Ministry of Supply, which has been associated with the development from its inception, maintains an active interest.

There is no suggestion that CO_2 cooling should be applied indiscriminately, but rather that it should be used only for operations on modern high-strength, tough and heat-resistant steels and alloys. With these materials production times can be reduced by 25 to 50 per cent and tool life materially lengthened. For a specific output of workpieces this extension of productive capacity may well mean a reduced capital expenditure on machine tools. Currently, a saving of one machine in four is claimed with confidence, but it is expected that eventually it will be possible to dispense with two of four machines. Conversely, of course, a substantially higher output can be obtained from existing tools without incurring heavy capital outlay.

*February 1954

Tests have indicated that in turning stainless steel, for example, the weight of stock removed may be at a rate five times that possible with conventional equipment and the working life of a tungsten carbide-tipped tool may be extended at least three times. These are substantial economies but there are also incidental advantages of some value. It is possible to use a much thinner carbide tip than is usually deemed necessary. Work produced is of more consistent quality since there appears to be no tendency for build-up to occur at the cutting edge of the tool. This factor, in conjunction with a high cutting speed and a hot chip, gives a markedly better surface finish to the work. Operations are clean and dry and the turnings or chips do not require degreasing or draining for coolant recovery.

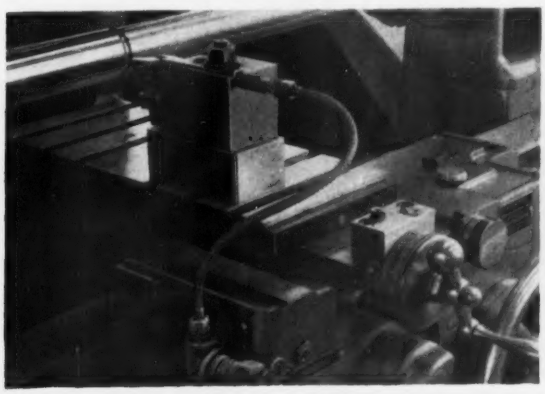


Standard tool, adapted for side entry of CO_2, cutting EN 25 steel on Fisher copying lathe

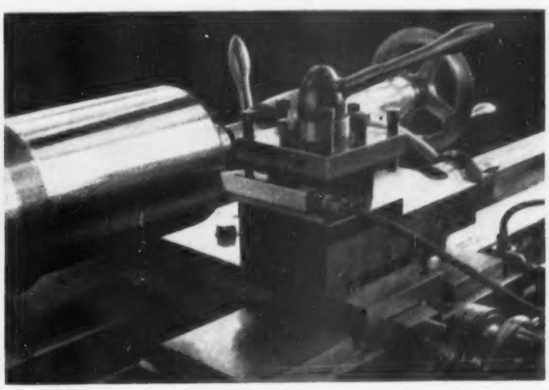
To obtain the best results the CO_2 jet is directed from below towards the cutting edge of the tool. Only the surplus reaches the workpiece and none passes to the chip. The rate of flow is determined by the final restriction in the flow line adjacent to the tool tip. An intermediate control embodying an orifice comparable in dimensions to that of the jet would occasion a drop in pressure resulting in the formation of solid CO_2 and interruption of the flow.

In the most recently developed method a reservoir of coolant is provided under the tool tip and the jet or jets are formed in the body of the tool below the tip. In the case of a tool inserted in a holder the CO_2 is fed to a recess in the tool socket and an orifice provided either through or around the side of the tool. Either adjustable or non-adjustable flow control devices are available which can be fitted in a boring in the tool shank. The first is a variable-orifice valve with direct screwed adjustment of the needle and the other a capillary tube of appropriate length mounted in the end of a support tube. For general use in the workshop the capillary tube is preferable since once the flow has been determined in the tool room it cannot be altered or maladjusted by the operator. In some instances it may be more convenient to secure a capillary tube jet in a groove machined in one side of the tool shank as shown, for instance, in the set-up of the Fisher copying lathe. For straightforward turning such tools can be used in the standard tool-post of an existing lathe and the only other equipment necessary is a length of flexible tubing and a shut-off cock mounted on a bracket attached at any convenient point.

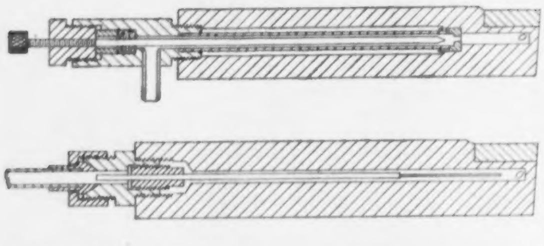
In drilling operations the CO_2 is



CO_2 cooled tool cutting Nimonic 90 alloy on H.E.B. lathe



Set-up of CO_2 cooled tool for normal bar turning



Variable-orifice or capillary flow control devices can be embodied in the tool shank

delivered in a jet impinging on the drill immediately above its entry into the work and heat at the cutting face is removed by conduction. Even when employing high speeds and feeds this dry application is satisfactory for holes of moderate depth. For deep holes it may be necessary to supply additionally oil to ensure removal of swarf. For end milling a similar jet is used to good effect since the distance from the cutting face to the point of CO_2 application is usually short. Surface milling may present a problem, particularly when large cutters are used, since to cool the complete tool is wasteful of CO_2 and it may be difficult to apply the coolant to the intermittently operative cutting edge. In such circumstances a small quantity of a light oil may be introduced into the jet to weight the CO_2 snow and enable it to be projected on to the cutting edge, to which it will adhere.

With either stationary or rotating boring bars the CO_2 coolant can be delivered at the tool tip and no difficulty is encountered where there is an adequate area for the discharge of swarf. For deep, close boring, however, it will be necessary to furnish an independent supply of oil or compressed air to carry away the swarf.

In tool grinding one or more CO_2 jets can maintain both the tool and the wheel at normal or sub-normal temperature. The wheel is left clean and dry and has no tendency to clog—there being no coolant residues. With the ordinary lathe tool the internally fitted capillary feed may be used during regrinding, either with or without supplementary surface cooling.

Carbon dioxide has a critical temperature of 31 deg C and below that temperature it can exist in the liquid state. The recommended method of delivering CO_2 to jets or capillaries is to operate at a pressure of 750 to 1,000 lb/in^2 and to lower its temperature to approximately 4 deg C on its way to the discharge points. These values make it possible to avoid difficult problems of refrigeration and the lagging of pipework. The operating pressure is considerably above the equilibrium pressure of the liquid at that temperature and thus there is no risk of premature expansion and consequent solidification before reaching the discharge point.

Machine	Material	Tool	Speed ft/min	Cut in	Feed in/rev	CO_2 lb/hr
Colchester lathe	EN 19	Wimet X8	300	0·010	0·004	5
H.E.B. lathe	Nimonic 90	Coromant Hl	190	0·020	0·022	5
Fisher lathe	EN 25	Cutanit S	475	0·100	0·014	5
Progress 4E drill	Stainless Steel	0·25 in. dia.	70	—	0·005	5
Victoria U2 mill	EN 2	H.S.S.	110	0·100	6·2 in/min	5

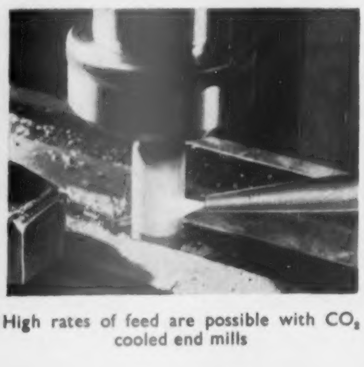
A small installation, adequate for one or two machine tools or for several small machines in a toolroom, comprises an eight-cylinder rack and a small electrically-operated refrigerator. By cooling the liquid between the storage cylinders and the jets the effect of ambient temperature is offset and virtually consistent standard conditions obtain. Each cylinder has a useful capacity of about 20 lb of liquid CO_2 and for convenient operation four are connected for supply through a common manifold. When these are emptied an alarm and an indicating signal are given and a permanently mounted reserve cylinder is brought into operation whilst the changeover to the four remaining full cylinders is made.

For production installations, bulk storage tanks automatically indicating their charge weight have been developed. In these, of 1·5 and 6·0 tons capacity respectively, the liquid CO_2 is maintained at -20 deg C and 290 lb/in^2 by a self-contained refrigerant system. Delivery from the tank is by means of a positive displacement pump which raises the pressure to 900 lb/in^2. When machines are sited close to the storage tank the cooled liquid can be delivered through lagged pipes but usually it is preferable to allow the liquid CO_2 to reach ambient temperature in unlagged pipes and to re-cool for groups of machines by means of small refrigerator units, as with the cylinder system. In a works equipped with a bulk storage system but using additionally rack installations for small remotely sited shops or isolated tools, the cylinders can be refilled from the bulk supply high pressure line. Stocks of CO_2 are maintained by the Carbon Dioxide Co. Ltd. at branches and depots throughout the British Isles. In most areas supplies can be delivered, either in cylinders or in bulk tank wagons, within 24 hr.

An indication of the results already obtained with CO_2 cooled tools in various applications is given by the examples listed. The performance figures are regarded as conservative and may be improved as the technique is further developed. As regards the cost of CO_2 cooling, a maximum figure of 6d. per lb for CO_2 delivered to the machine may be estimated, it is stated.

Differing from a number of refrigerants, carbon dioxide is non-toxic. It has

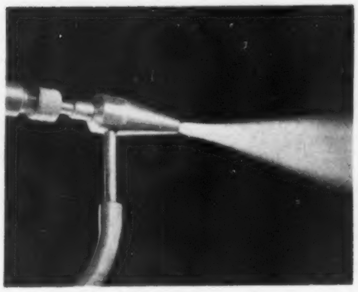
Dry-drilling stainless steel with CO_2 coolant



High rates of feed are possible with CO_2 cooled end mills

Jet cooling of tool and wheel in off-hand tool grinding

been established that the presence of CO_2 in the atmosphere up to a concentration of 5 per cent has no deleterious physiological effect. When used as a tool coolant it has to be considered merely as an addition to the shop atmosphere. Its use involves no loss of oxygen and no production of impurities such as occurs, for example, on the combustion of carbonaceous material when the formation of carbon dioxide is accompanied by the removal of an equivalent amount of oxygen, and some carbon monoxide may be produced. Since the recommended standard of ventilation requires six changes of air per hour there is virtually no possibility of a



Oil injection nozzle for projection of CO_2 snow to awkward locations

concentration of more than approximately 1·0 per cent being reached in a shop in which all machines are equipped for CO_2 cooling. As carbon dioxide is more dense than air there will be a tendency to collect at floor level under still air conditions. In the average shop the general disturbance of the air by moving parts of machines and the movement of operators renders such separation extremely improbable. Should a concentration occur, for instance adjacent to a machine using a heavy discharge of CO_2, its presence can be detected by inexpensive instruments and it can be dispersed by the simplest means.

ENGINE DEPOSITS

AN *S.A.E. Preprint* entitled "Combustion Chamber Deposits and Octane Number Requirements," by J. Warren, January 11-15, 1954, is now available. In it the author states that elimination of the effects of combustion chamber deposits on the anti-knock requirements of engines would, in the current octane number range, increase the fuel utilization efficiency by some 5 per cent. Laboratory tests are reported in which particular attention was given to the question of whether the thermal effects, which contribute a large part of the total effect of deposits, result from thermal insulation or heat capacity.

A single cylinder, CFR knock-test engine with an L-head cylinder was specially modified to permit measurement of heat transfer during deposit accumulation. The fuel used was iso-octane. Arrangements were made to determine the increase in compression ratio that occurs as deposits accumulate. This increase influences the anti-knock requirements of the engine. A table gives details of the operating conditions used in the tests. Air-flow was measured by a flow-meter installed ahead of a control valve in the air line to the carburettor.

Preliminary investigations showed that the sum of the thermal and volume effects accounts for practically all the increase in octane requirement caused by deposit accumulation. Exclusive of volume effects, accumulated deposits may raise the end-gas temperature of the charge, and hence the octane number requirement, by one or more of the following methods: (1) absorbing heat from one cycle and transferring it to the incoming charge during the following suction stroke, (2) heating the charge during the compression stroke, (3) acting as thermal insulation for part of the compression stroke and during combustion, and thereby reducing the heat dissipated from the charge to the coolant. Experiments relating to engine air consumption and octane number requirement pointed to the deposit thermal effects as resulting from the heat capacity of the deposits. Further tests were made on deposit effects and on factors possibly influencing the relationship between thermal insulation and heat capacity effects.

It was concluded that deposit thermal effects are due to the heat capacity of the deposits, the incoming charge being heated by the energy stored in the deposits from the previous cycle. Depending on fuel type and engine operating conditions, thermal effects account for 70-100 per cent of the increase in octane number requirements, and volume effects for 0-30 per cent. Engines with the smallest area of exposed combustion chamber surface for a given displacement would be expected to have the smallest thermal effects and hence minimum deposit effects. *M.I.R.A. Abstract No.* 6720.

PISTONS AND RINGS

IN an article entitled "Pistons and Rings for High Output Diesel Engines," by J. W. Pennington, in *Gas Oil Power, Annual Technical Review Number*, 1953, *Vol.* 48, *No.* 581, the principal problems encountered in pistons are stated to be as follows: structural failures in the crown, or in or above the gudgeon pin bosses, deterioration of ring grooves, and deposits in the ring belt. Materials may be cast iron, high-strength iron, steel alloy, or aluminium. The light metal, aluminium, allows a reasonably rigid section to be used, but has low resistance to ring-groove wear under severe operating conditions. On the other hand, cast iron or steel pistons have given little trouble in this respect. Inserts of austenitic or grey iron, either bonded or unbonded, at the ring belt have been found satisfactory with aluminium pistons, centrifugally cast inserts being least liable to distortion on machining. Determination of the location of the weak spots in pistons has been helped by the use of stress coats. Ring sticking is affected mainly by the fuel, the lubricating oil, and the piston temperature.

The basic piston ring problems are: obtaining satisfactory blow-by control, oil control and life, which means resistance to breakage, wear and sticking. These requirements sometimes operate against one another; for example, narrow rings help to prevent scuffing, but increase the chance of fracture. High strength materials reduce breakage, but have poorer surface running characteristics. Blow-by control is basically the same for two-stroke and four-stroke engines, although in two-stroke units it may be necessary to incorporate compression or sealing rings which traverse the ports.

Two main factors affect oil control. They are: unit pressure against the cylinder wall, and conformity. Numerous types of oil control ring are described. In addition, the need is shown for a design having a ring face that will not increase appreciably in width as the ring wears, and with a low spring rate so that radial pressure remains substantially constant with wear. With ordinary cast iron material, designs incorporating maximum tension and minimum practical face width are the most effective, but if high strength, high modulus materials, such as the malleable irons, are used, the face width can be narrower without risk of breakage, but conformity is reduced.

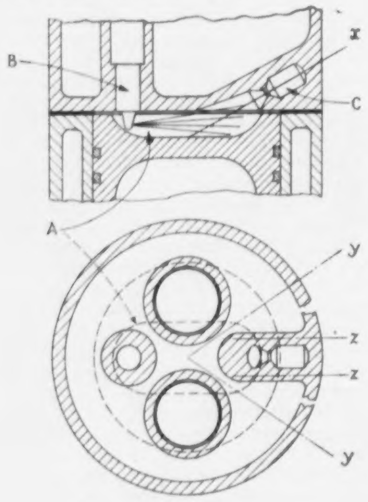
Rings with expanders, conformable oil control rings, and various types of plated, grooved, tapered and inserted-faced rings are described. The author is not entirely in favour of torsionally twisted, taper-face compression rings. *M.I.R.A. Abstract No.* 6695.

CURRENT PATENTS

A Review of Recent Automobile Specifications

Air-cell type combustion chamber

THIS arrangement of the combustion chamber of an air-cell type diesel engine is designed to ensure that sufficient fuel enters the cell to produce a discharge adequate to promote in the working cylinder the turbulence necessary to obtain complete combustion. Formed entirely in the crown of the piston, the combustion space A is either trough-shaped, as shown, or elliptical.



No. 704024

Fuel injector B and air cell C are disposed in a plane normal to the medial plane of a multi-cylinder unit and containing the major axis of the combustion chamber. Mounted with its axis parallel to the cylinder axis, the injector protrudes beyond the face of the cylinder head so that the lateral spray orifice is positioned at about half-depth of the combustion space. The cylindrical air cell is arranged at an inclination on the opposite side of the cylinder head with its axis *x* meeting the cylinder axis at the base of the combustion space when the piston is at top dead centre position.

In modifications of the arrangement, two air cells of correspondingly smaller capacity are provided instead of a single cell. They may be disposed on axes *y, y* or *z, z* inclined as before to the base of the combustion chamber. The conical discharge orifices of such a pair of cells merge to form a single depression in the cylinder head against which the fuel is directed. *Patent No.* 704024. *Lanova A.G.* (*Switzerland*).

Plain bearings

IMPROVED resistance to break-up by fatigue cracking of the facing layer is claimed for this method of producing electro-deposited thin-layer, steel-backed bearings. The triplex bearing structure comprises a low carbon steel backing, an interlayer of copper bonded to the backing, and a thin layer of soft bearing metal electro-deposited on the interlayer. According to the invention, the copper is electro-deposited on the backing from a suitable plating solution under such conditions that a crystalline structure presenting a creviced or acicular surface is produced. The soft metal layer is electro-deposited on this rough surface and the resulting bond has a mechanical strength additional to the metallurgical adhesion.

The formation of the creviced surface depends mainly on the freedom from impurities, particularly organic matter, in the plating solution, but is influenced by the cleaned and etched surface of the backing. It can be modified by fine scratch brushing of the backing, which will encourage the formation of crevices, or by the introduction of an organic agent such as cresol sulphonic acid should the surface become too coarse.

The thicknesses of the respective layers will vary according to the size, loading and other factors of the bearing, but the invention refers, non-exclusively, to backings of 0·030 to 0·100 in thick, interlayers of 0·002 to 0·015 in thick and facing layers of 0·0005 to 0·002 in thick. *Patent No.* 704085. *Glacier Metal Co. Ltd.*

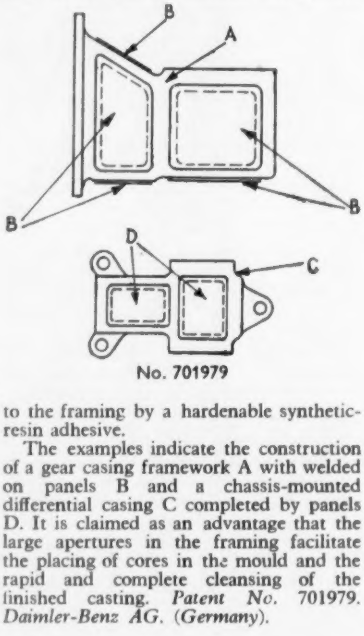
Boot lid counterbalance

A PAIR of torsion rods is employed in this hinge arrangement to counterbalance the weight of a luggage boot lid. The lid is carried by two cranked levers A pivoted at B on brackets C secured inside the luggage compartment. Pivoted on each lever at D is a link E, the free end engaging the crooked end of torsion rod F which extends also through an arcuate guide slot in the bracket. The torsion rod is supported in the bracket at G and is anchored by a lug H on the bracket at the other side of the compartment.

When the lid is shut down the moment of its weight is sufficiently greater than the moment exerted by the torsion rods and it remains closed. As the lid is swung upwards, requiring only light effort, the moment of its weight is reduced to a greater extent than the stress in the torsion rods is reduced and the lid is held in the open position. *Patent No.* 704399. *Wilmot-Breeden, Ltd.*

Composite construction of gear housings

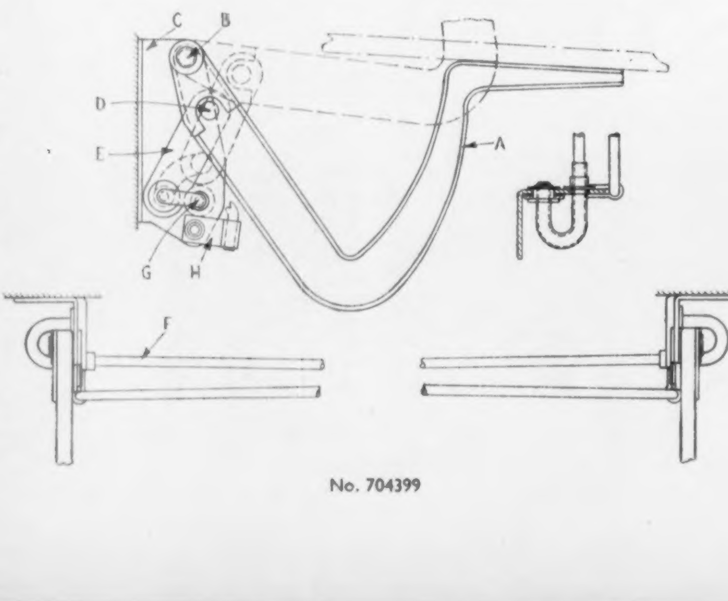
FOR the housings of components such as change-speed gearboxes or de Dion-type rear axle units, weight is saved by using a cast skeleton framework to carry all stresses and completing the enclosure by irremovable thin sheet panels. Any suitable metal, ferrous or non-ferrous, may be used for the panels which may be cast-in the structure or welded on the cast framework. Alternatively, the panels may be of plastics material and secured



No. 701979

to the framing by a hardenable synthetic-resin adhesive.

The examples indicate the construction of a gear casing framework A with welded on panels B and a chassis-mounted differential casing C completed by panels D. It is claimed as an advantage that the large apertures in the framing facilitate the placing of cores in the mould and the rapid and complete cleansing of the finished casting. *Patent No.* 701979. *Daimler-Benz AG.* (*Germany*).



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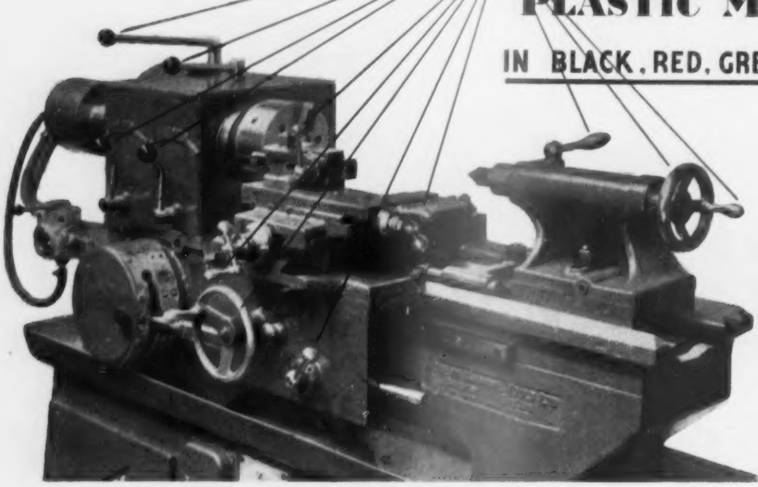

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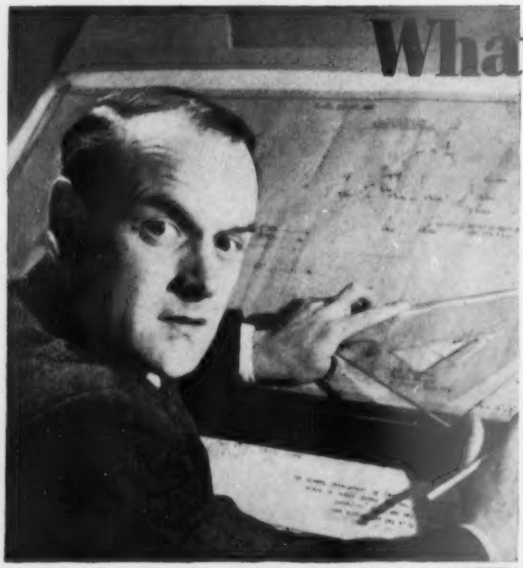

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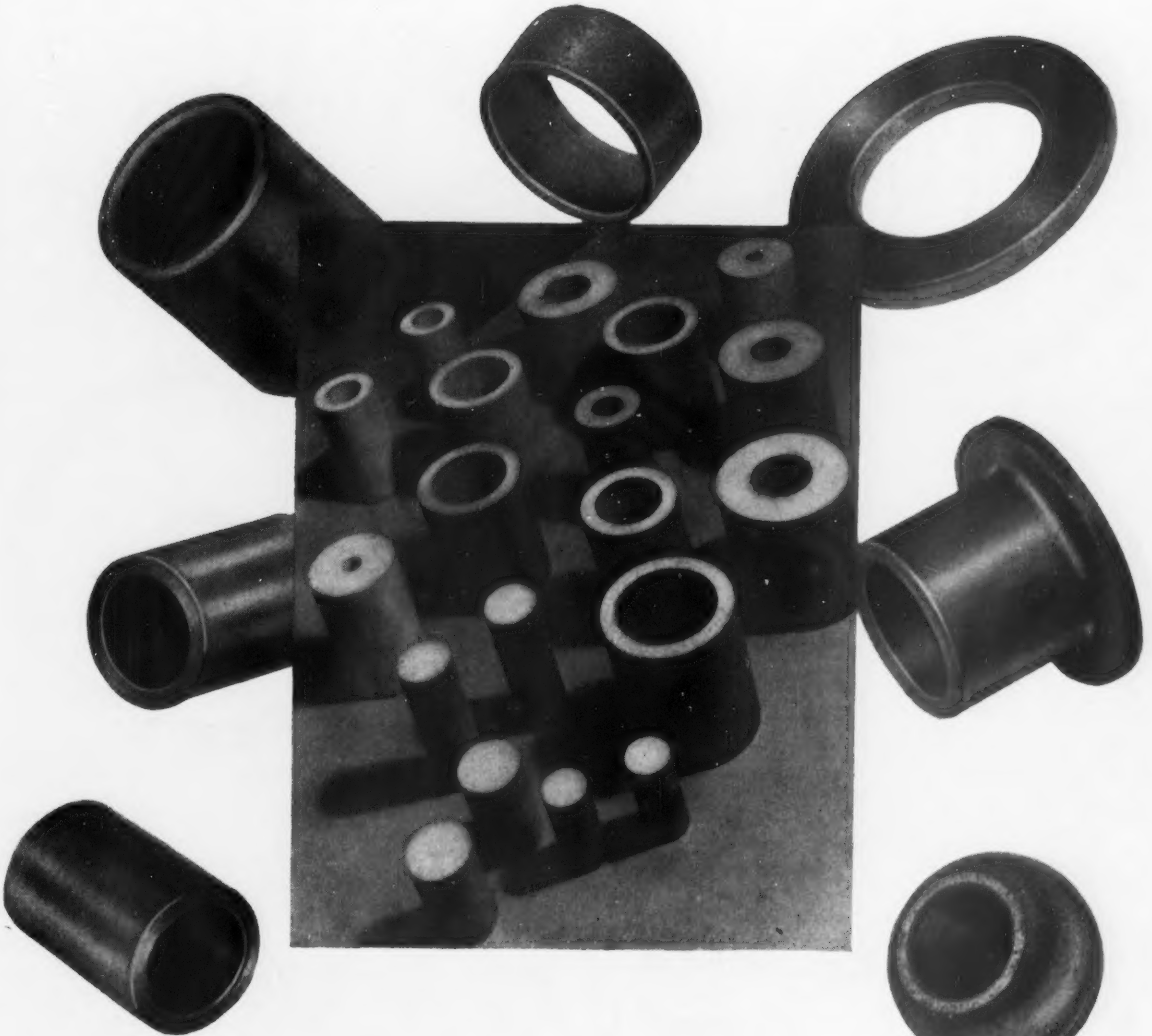

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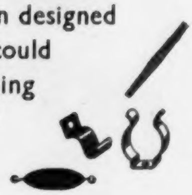

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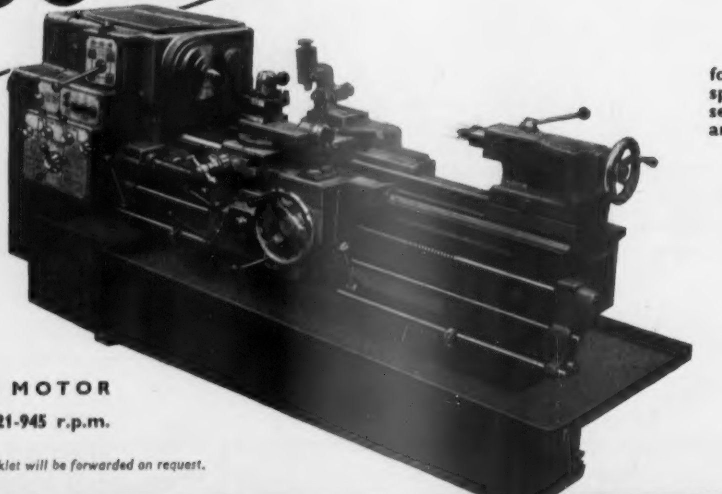

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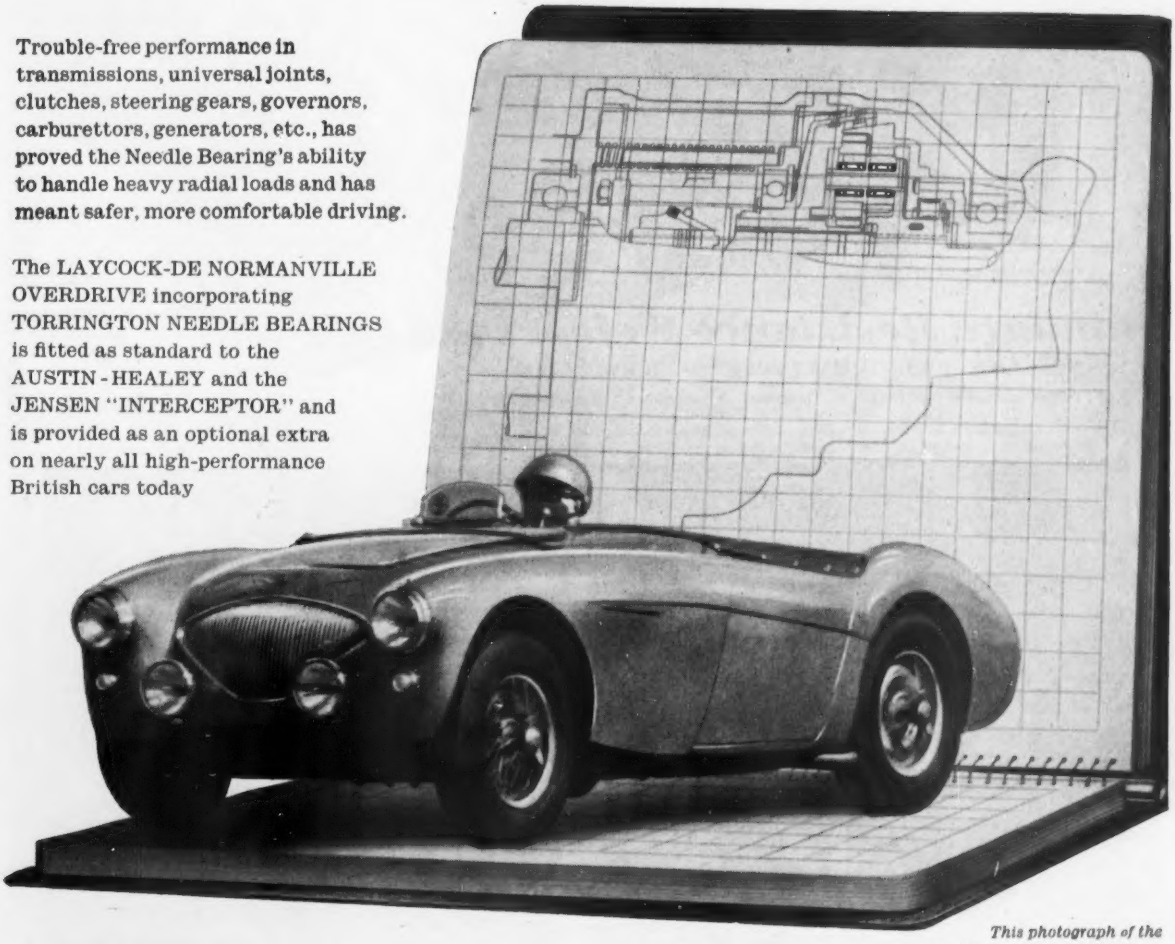
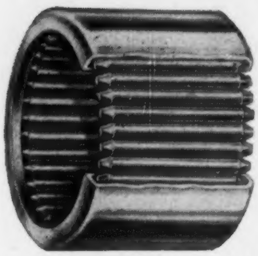


This photograph of the Austin-Healey record-breaking car is reproduced by kind permission of The Austin Motor Co. Ltd.



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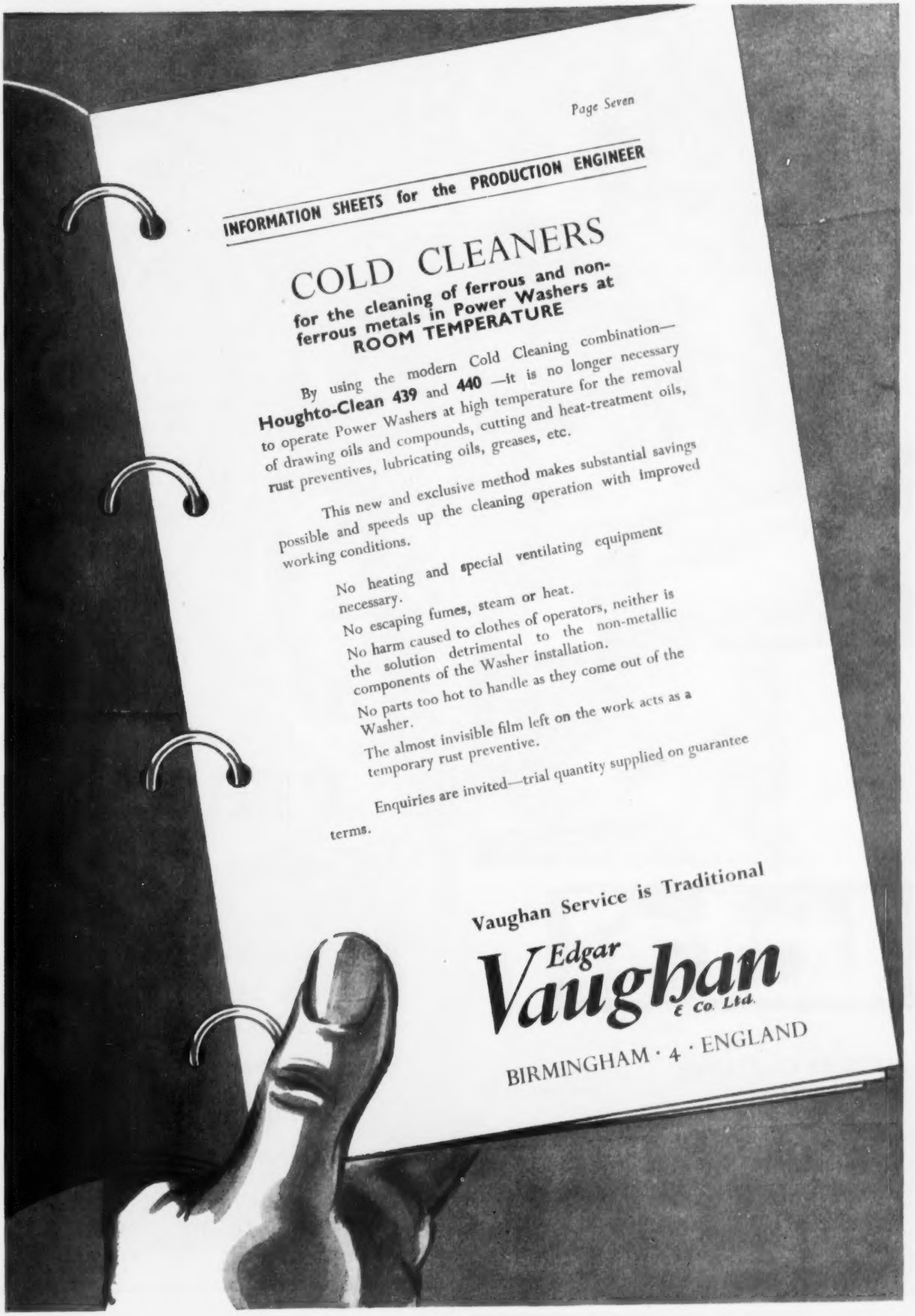
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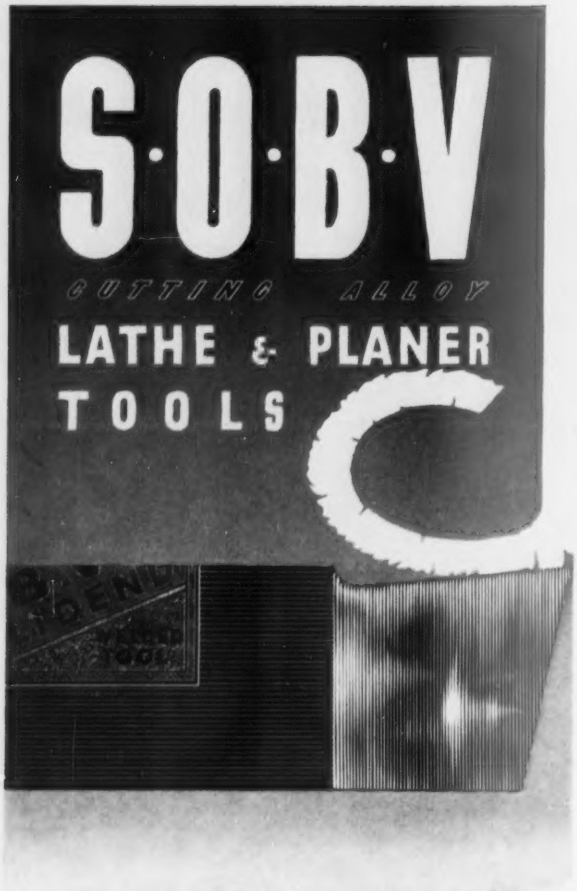

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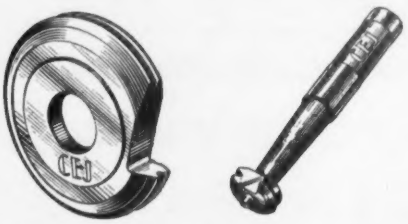

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APPLICATION in writing to Secretary, A.E.C. Ltd., Windmill Lane, Southall, Middlesex. [4820

A LEADING Automobile Manufacturer in the Midlands has a vacancy for a Senior Design Engineer. He is required to take charge of petrol engine design for a range of cars and commercial vehicles. Applicants, who must be familiar with modern developments and production techniques, should state fully their age, qualifications and experience. Reply to Box 5627 c/o Automobile Engineer. [4824

DESIGNER - DRAUGHTSMAN — Section Leader status—with design experience of small hydraulic cylinders and controls, for development and application of equipment for the heavy automobile and similar fields. Knowledge of road vehicles an advantage. Also competent Draughtsman for similar work. Write in confidence, with full details of qualifications, experience, personal particulars and salary required, to Chief Engineer, Clayton Dewandre Co., Ltd., Titanic Works, Lincoln. [4818

SITUATIONS VACANT

APPLICATIONS are invited from qualified personnel for the following positions in our Engineering organisation engaged on the development of Fuel Injection Equipment for Diesel Engines.

ENGINEERS with experience of high speed diesel engines and/or fuel injection equipment for supervision of research and development and engine testing and application of fuel injection equipment to vehicle, tractor and small industrial engines. Degree standard or H.N.C. desirable.

SENIOR and Junior Draughtsmen. Preferably with experience of small internal combustion engines or light mechanical work.

JUNIOR Technical Assistants and Engine Testers accustomed to dynamometer testing of internal combustion engines.

PLEASE apply giving full details to: Personnel Manager, Simms Motor Units, Ltd., Oak Lane, Finchley, N.2. [4823

SENIOR Technical Assistants required for interesting development work to at least H.N.C. standard. Specialised knowledge in one of the following an advantage. Salary range £600-£800 per annum.

(1) Transmission.
(2) Engine installations on test bed and in vehicles.
(3) Turbines.
(4) Thermionics.
(5) General vehicle development.

FACTORY situated on southern outskirts of Birmingham. Please apply Box 5585 c/o Automobile Engineer. [4822

RESEARCH and Development Worker required in the Automobile Development Dept. Degree, Higher National Certificate or equivalent. Write stating training experience and qualifications. Personnel Dept., Austin Motor Co., Ltd., Longbridge Works, Birmingham. [4821

SENIOR and Intermediate Designer/Draughtsmen required for design and detail of vehicle component assemblies. Applicants having good general light engineering training and experience will be considered. Interesting range of work, excellent working conditions and rates. Pension scheme. 5-day week. Apply in writing giving full particulars to Chief Engineer, Clayton Dewandre Company Limited, Titanic Works, Lincoln. [4819

DRAUGHTSMEN (Junior and Senior) required, preferably with automobile experience. Harry Ferguson Research Ltd., Fletchamstead Highway, Coventry. [4826

LONDON County Council. Paddington Technical College—September 1954. Grade "A" Assistant to teach Motor Vehicle Technology and allied subjects, mainly to City and Guilds Motor Vehicle Classes. Salary within Burnham scale—£451-£736; commencing salary and maximum according to age, qualifications and experience. Application forms from Secretary at College, Saltram Crescent, W.9, for return by 25th June (693). [4827

DESIGNERS: Excellent opportunity for first-class men. Preference given to applicants with experience in Cars, Commercial Vehicles and Railway Rolling Stock, but good general engineering experience acceptable. Essential to have University degree or equivalent engineering qualifications. Good salary, modern offices, staff pension scheme, Birmingham area. Applications to Personnel Manager. Box 5752, c/o Automobile Engineer. [4829

SITUATIONS WANTED

AUTOMOBILE Engineer (Diploma), 24, all-round experience commercial and technical, market research abroad (hand and electrical tools), service literature and translations, fluent Swedish, French, German, seeks progressive situation, home or overseas. Box 5713, c/o Automobile Engineer. [4628

WANTED

COLLECTOR wants old car catalogues, etc.—Harris, 4, Crawford Road, Wolverhampton. [4817

BOOKS

OXYGEN Cutting: A Comprehensive Study of Modern Practice in Manual and Machine Cutting. By E. Seymour Semper, M.I.Mech.E., M.Inst.W. Written to assist engineers concerned with cutting and shaping material, this book describes many of the machines designed for various applications of oxygen cutting to template with multiple heads and also the actual methods of operation. Price 10s. 6d. net. By post 10s. 11d. From all booksellers or from The Publishing Dept., Dorset House, Stamford Street, London, S.E.1.

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